GTU Study Materials ☑

Syllabus
B.E Semester : VII (NEW)
(Mechanical Engineering)
Subject Code 2171910 Subject Name : Power Plant Engineering

1 **Thermal Power Plant:** General layout of modern thermal power plant, Siteselection, Presents status of power generation in India

2 **High Pressure Boilers:** (Unique features and advantages of high pressureboilers, LaMont; Benson; Loeffler and Schmidt-Hartmann boilers)*, supercritical boilers, Supercharged and fluidized bed combustion, Methods of superheat control, Corrosion in boilers and its prevention

3 **Coal and Ash Handling Systems:** Coal storage, Burning systems, Types ofstokers and their working, Pulverized fuel handling systems, Unit and central systems, Pulverized mills- ball mill, Bowl mill, Ball & race mill, Impact or hammer mill, Pulverized coal burners, Oil burners, Necessity of ash disposal, mechanical; hydraulic; pneumatic and steam jet ash handling system, Dust collection and its disposal, Mechanical dust collector, Electrostatic precipitator

4 **Draught System:** Natural draught – estimation of height of chimney,Maximum discharge condition, Forced; induced and balanced draught, Power requirement by fans

5 **Steam Nozzles:** Types of nozzles, velocity of steam, discharge through nozzle,critical pressure ratio and condition for maximum discharge, physical significance of critical pressure ratio, nozzle efficiency

6 **Steam turbine:** (Principle of operation, types of steam turbines, compoundingof steam turbines, impulse turbine – velocity diagram)*, calculation of work, power and efficiency, condition for maximum efficiency, Reaction turbines – velocity diagram, degree of reaction, reheat factor, (governing of steam turbine– throttle, nozzle and bypass governing)*, Methods of attachment of blades to turbine rotor, Labyrinth packing, Losses in steam turbine

7 **Condensers and Cooling Towers:** Types of condensers, sources of air incondenser, Effects of air leakage, Methods of obtaining maximum vacuum in condenser, vacuum & condenser efficiency, Mass of cooling water required,Edward air pump, Necessity of cooling ponds and cooling towers, Condenser water cooling systems, Types of cooling towers and cooling ponds

8 **Feed Water Treatment:** Necessity of feed water treatment, Different impuritiesfound in feed water, Effect of impurities, pH & its role in corrosion and scale formation,

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Internal & external water treatment systems – Hot lime soda process, Zeolite ion exchange process, Demineralization plants, Reverse osmosis process, Sea water treatment using reverse osmosis, De-aeration

9 **Gas turbine:** Classification, open and closed cycle, gas turbine fuels, actualBrayton cycle, optimum pressure ratio for maximum thermal efficiency, work ratio, air rate, effect of operating variables on the thermal efficiency and work ratio and air rate, combined steam and gas turbine plant, gas turbine blade cooling

10 **Nuclear Power Plant:** Nuclear fusion and fission, Chain reaction, Nuclear fuels,Components of nuclear reactor, Classification of reactors, Pressurized water reactor, Boiling water reactor, Gas cooled reactor, CANDU reactor, Fast breeder reactor, Nuclear waste and its disposal, Nuclear power plants in India

11 **Jet Propulsion:** Turbojet Engine*, thrust, thrust power, propulsive efficiency,thermal efficiency, (Turboprop, Ramjet and Pulsejet engines, Rocket engines)*

12 **Economics of Power Generation:** Load curves, Load duration curves,Connected load, Maximum load, Peak load, Base load and peak load power plants, Load factor, Plant capacity factor, Plant use factor, Demand factor, Diversity factor, Cost of power plant, Performance and operating characteristics of power plant, Tariff for electric energy.

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Preface to the Fifth edition

The authors feel very pleased to bring out the Fifth edition, which has been thoroughly revised as per new syllabus for 7th semester effective from June '16.

The authors are grateful to the teachers and students of various engineering institutions for their appreciation and healthy criticism of earlier editions of the book.

We express our sincere thanks to our colleagues, students and the teachers of different engineering colleges for their valuable suggestions and recommending the book of their students.

Readers both students and teachers are requested to send their comments. Constructive suggestions for further improvement are most welcome and would be incorporated in the next edition.

Suggestions for the further improvement of text will be highly appreciated and thankfully acknowledged.

June, 2016

H. G. Katariya
J. P. Hadiya

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Contents

Chapter 1 Thermal Power Plant

1.1 Introduction 1
1.2 Classification of power plant 1
1.3 Thermal power plant 3
1.4 General layout of modern thermal power plant 6
1.5 Site selection of thermal power plant 10
1.6 Present status of power generation in India 12
1.6.1 Energy sources and present power installed capacity in India 12
1.6.2 Power crisis in India 24
1.6.3 Per capita annual power consumptin of India 25
1.6.4 Role of public sectors in development of power in India 25
1.7 Comparisions of various power plants 34
• Exercise. 36
• Objective Type Questions 36

Chapter 2 High Pressure Boilers

2.1 Introduction to high pressure boilers 39
2.2 Unique features of high pressure boilers 40
2.3 La-Mont boiler 43
2.4 Benson boiler 44
2.5 Loeffler boiler 46
2.6 Schmidt-Hartmann boiler 47
2.7 Supercritical boiler 49
2.8 Supercharged boiler 51
2.9 Fluidized Bed Combustion (FBC) system 52
2.9.1 Atmospheric FBC boilers 55
2.9.2 Pressurized FBC boiler 57
2.10 Heating surface and their location in the boiler 58
2.11 Superheaters 59
2.12 Reheaters 63
2.13 Economizers 64
2.14 Air pre-heaters 67
2.15 Steam temperature or superheat control 70
2.16 Corrosion in boilers and its prevention 74
• Exercise 84
• Objective Type Questions 86

Scanned with CamScanner

CamScanner

Chapter 3 Coal and Ash handling Systems

3.1 Introduction 90
3.2 Out-Plant handling of coal 90
3.3 Storage of coal 92
3.3.1 Dead storage or outdoor storage 92
3.3.2 Live storage or active storage 93
3.4 In-plant handling of coal 94
3.4.1 Coal delivery 94
3.4.2 Unloading 95
3.4.3 Preparation of coal 95
3.4.4 Transfer of coal 96
3.4.5 Inplant handling 100
3.4.6 Weighing of coal 101
3.5 Coal Burning System 101
3.6 Solid fuel firing 102
3.6.1 Hand firing 102
3.6.2 Mechanical firing (stoker firing) 102
3.7 Types of stoker firing 102
3.7.1 Principle of overfeed stokers 102
3.7.2 Principle of underfeed stokers 106
3.8 Pulverized fuel system 110
3.9 Pulverized fuel handling systems 112
3.9.1 Unit system 112
3.9.2 Central (Bin) System 113
3.10 Pulverizing mills 114
3.10.1 Ball mill 114
3.10.2 Ball and race mill 116
3.10.3 Bowl mill 117
3.10.4 Hammer (impact) mill 118
3.11 Pulverized coal burners 119
3.11.1 Long flame or U-flame or streamlined burner 119
3.11.2 Short flame or turbulent burner 120
3.11.3 Tangential burners 121
3.11.4 Cyclone burner 122
3.12 Oil burners 123
3.13 Necessity of ash disposal 124
3.14 Ash handling systems 125
3.15 Mechanical ash-handling system 126
3.16 Hydraulic ash-handling system 127
3.16.1 Low-velocity system 127
3.16.2 High-velocity system 127
3.17 Pneumatic ash handling system 128
3.18 Steam jet ash handling system 129

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CamScanner

3.19 Dust collection and its disposal 130
3.20 Mechanical dust collector 130
3.20.1 Gravitational separators 130
3.20.2 Cyclone separator 132
3.20.3 Wet type mechanical dust collector 133
30.21 Electrostatic precipitator (ESP) 133
• Exercise 136
• Objective Type Questions 137

Chapter 4 Draught System

4.1 Introduction to draught 141
4.2 Losses in the air-gas circuit of boiler 141
4.3 Classification of draught 143
4.4 Natural draught 143
4.4.1 Estimation of height of chimney 144
4.4.2 Draught in terms of hot gases column 146
4.4.3 Diameter of chimney 146
4.4.4 Condition for maximum discharge through a chimney 147
4.4.5 Draught at maximum discharge 148
4.4.6 Efficiency of chimney 148
4.4.7 Advantages and limitations of chimney or natural draught 149
4.5 Mechanical draught 159
4.5.1 Forced draught 160
4.5.2 Induced draught 160
4.5.3 Balanced draught 161
4.5.4 Comparision between forced and induced draught 162
4.5.5 Power requirement by fans 163
4.6 Advantages and disadvantages of mechanical draught over natural draught 170
4.7 Steam jet draught 171
• Exercise 172
• Objective Type Questions 174

Chapter 5 Steam Nozzles

5.1 Introduction 177
5.2 Types of nozzles 177
5.3 Velocity of steam flowing through a nozzle 178
5.4 Friction in a nozzle or Nozzle efficiency 180
5.5 Mass flow rate of steam through the nozzle 181
5.6 Condition for maximum discharge through a Nozzle or Critical pressure ratio or choked flow 183
5.7 Physical significance of critical pressure ratio 185
5.8 General relationship between area, velocity and pressure 195

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CamScanner

5.9 Supersaturated flow in nozzles 207
5.10 Effect of variation of back pressure 209
• Exercise 219
• Objective Type Questions 221

Chapter 6 Steam Turbine

6.1 Introduction 225
6.2 Principle of operation 226
6.3 Classification of steam turbines 227
6.4 Compounding of steam turbines 231
6.4.1 Pressure compounded impulse turbine (Rateau Turbine) 231
6.4.2 Velocity compounded impulse turbine (Curtis turbine) 233
6.4.3 Pressure -Velocity compounded of impulse turbine 234
6.5 Impulse turbine - velocity diagram 235
6.5.1 Calculations for work power and efficiencies 237
6.6 Condition for maximum blade efficiency of impulse turbines 238
6.7 Reaction Turbine 253
6.8 Velocity diagram for reaction turbine 253
6.9 Degree of reaction 254
6.10 Parson's turbine 255
6.11 Condition for maximum efficiency of Parson's reaction turbine 256
6.12 Reheat-factor 280
6.13 Internal and other efficiencies 282
6.14 Governing of steam turbines 288
6.14.1 Throttle governing 289
6.14.2 Nozzle governing 291
6.14.3 By-pass governing 291
6.14.4 Comparison of throttle and nozzle governing 292
6.15 Methods of attachment of blades to turbine rotor 293
6.16 Labyrinth packing 297
6.17 Losses in steam turbines 299
6.17.1 Internal losses 299
6.17.2 External losses 303
6.18 Special types of steam turbines 303
6.19 Back pressure turbines 303
6.20 Pass out turbine or Extraction turbine 305
6.21 Mixed pressure turbine 307
• Exercise 314
• Objective Type Questions 318

force × induced thrust

Scanned with CamScanner

CamScanner

Chapter 7 Condensers and Cooling Towers

7.1 Introduction 321
7.2 Types of condensers 323
7.3 Jet condensers 323
7.3.1 Low level-parallel flow jet condenser 324
7.3.2 Low level-counter flow jet condenser 324
7.3.3 High level jet condenser 325
7.3.4 Ejector condenser 326
7.3.5 Advantages and disadvantages of jet condensers 327
7.4 Surface condensers 328
7.4.1 Double pass surface condenser 328
7.4.2 Down flow surface condenser 329
7.4.3 Central flow surface condenser 329
7.4.4 Evaporative condenser 330
7.4.5 Advantages and disadvantages of surface condensers 331
7.4.6 Comparison between jet and surface condensers 331
7.4.7 Requirements of a good surface condenser 332
7.5 Sources of air in condenser 332
7.6 Effects of air leakage 332
7.7 Methods of obtaining maximum vacuum in condenser 332
7.8 Edward air pump 333
7.9 Dalton's law of partial pressure 334
7.10 Vacuum and condenser efficiency 335
7.11 Mass of cooling water required for condensation of steam 336
7.12 Necessity of cooling ponds and cooling towers 355
7.13 Condenser water cooling systems 355
7.14 Types of cooling towers 358
7.14.1 Natural draught cooling towers 359
7.14.2 Forced draught cooling tower 360
7.14.3 Induced draught counter flow tower 361
7.14.4 Induced draught cross-flow Tower 361
7.14.5 Comparison between natural and mechanical draught cooling tower 363
7.15 Cooling ponds 363
• Exercise 366
• Objective Type Questions 368

Chapter 8 Feed Water Treatment

8.1 Necessity of feed water treatment 371
8.2 Different impurities found in feed water 372
8.3 Effect of impurities 374
8.4 pH and its role in corrosion and scale formation 376
8.5 Feed water treatment 378

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CamScanner

8.6 Internal Water treatment systems 379
8.7 External water treatment systems 381
8.8 Hot lime soda process 381
8.9 Zeolite or Ion exchange process 384
8.10 Demineralization plant 385
8.11 Reverse Osmosis process 387
8.12 Sea water treatment using reverse osmosis 388
8.13 De-aeration 389
• Exercise 391
• Objective Type Questions 391

Chapter 9 Gas Turbines

9.1 Introduction 393
9.2 Basic working principal of gas turbine 394
9.3 Advantages and disadvantages of gas turbine 395
9.4 Applications of gas turbines 397
9.5 Classification of gas turbines 397
9.6 Simple open cycle gas turbines 399
9.7 Closed cycle gas turbine 401
9.8 Comparison between the open cycle and closed cycle gas turbine 403
9.9 Gas turbine fuels 404
9.10 Ideal Brayton cycle 405
9.11 Work ratio and air rate 409
9.12 Actual Brayton cycle 412
9.13 Optimum pressure ratio for maximum work output 415
9.14 Optimum pressure ratio for maximum thermal efficiency 417
9.15 Effect of operating variables on thermal efficiency 437
9.16 Effect of operating variables on air rate 441
9.17 Effect of operating variables on work ratio 444
9.18 Means of improving efficiency and specific work output of simple open cycle gas turbine 446
9.19 Effect of various modifications 447
9.20 Water injection 450
9.21 Combustion chambers 455
9.22 Combined steam and gas turbine power plant 458
9.23 Different arrangement of combined cycle power plant 461
9.24 Thermodynamic analysis of combined cycle 463
9.25 Gas Turbine blade cooling 487
• Exercise 491
• Objective Type Questions 494

Scanned with CamScanner

CamScanner

Chapter 10 Nuclear Power Plant

10.1	Introduction	497
10.2	Nuclear fusion and fission	498
10.3	Chain reaction	501
10.4	Nuclear fuels	503
10.5	Components of nuclear reactor	504
10.6	Classification of reactors	507
10.7	Pressurized water reactor	508
10.8	Boiling water reactor	509
10.9	Gas cooled reactor	510
10.10	CANDU reactor	512
10.11	Liquid metal-cooled reactor	513
10.12	Fast breeder reactor	515
10.13	Nuclear waste and its disposal	516
10.14	Site selection for nuclear power plant	518
10.15	Nuclear power plants in India	518
10.16	Advantages and disadvantages of nuclear power plant	519
• Exercise		520
• Objective Type Questions		521

Chapter 11 Jet Propulsion

11.1	Introduction	525
11.2	Principle of Jet Propulsion	525
11.3	Classification of propulsive engines	526
11.4	Ram jet	528
11.5	Pulse jet engine	530
11.6	Turbojet	531
11.7	Thrust, Thrust power, Propulsive efficiency and Thermal efficiency	535
11.8	Turbojet engine with after burner	538
11.9	Turboprop	539
11.10	Merits and demerits of jet propulsion over other system	540
11.11	Rocket engine	556
11.12	Basic theory of operation of rocket engines	556
• Exercise		562
• Objective Type Questions		564

Chapter 12 Economics of Power generation

12.1	Introduction	567
12.2	Load curves	567
12.3	Load duration curves	569

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• 12.4	Terms and definitions	570
• 12.5	Importance of Load factor and diversity factor	574
• 12.6	Base load and peak load power plants	576
• 12.7	Economic load sharing between base load and peak load power stations	577
• 12.8	Cost of power plant	578
• 12.9	Performance and operating characteristics of power plant	582
• 12.10	Tariff for electric energy	584
• Exercise		611
• Objective Type Questions		614

• Papers 617

May '16
Dec. '15
May '15
Dec. '14
June '14
Dec. '13
May '13
Jan. '13
June '12
Nov.-Dec. '11
June '11
Dec. '10

TE Papers

May '16
Dec. '15
May '15
Nov. '14
May '14
Sep. '13
May '13
May '12

• Appendix - I	A659-A666
• Appendix - II	A667-A668
• Appendix - III	A669-A672
• Conversion Factors	A673-A674
• Molier Chart	

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Chapter 1

Thermal Power Plant

1.1 Introduction to power plant
1.2 Classification of power plant
1.3 Thermal power plant
1.4 General layout of modern thermal power plant
1.5 Site selection of thermal power plant
1.6 Present status of power generation in India
1.6.1 Energy sources and present power installed capacity in India
1.6.2 Power crisis in India
1.6.3 Per capita annual power consumption of India
1.6.4 Role of public sectors in development of power in India
1.7 Comparisions of various power plants

1.1 Introduction to power plant

Power plant is an assembly of equipments that produces and delivers mechanical and electrical energy. The electrical energy occupies the top position in the energy grades ranking. It finds unaccountable uses in home, industry, agriculture and even in transport. The electrical energy is a convenient form of energy because it can be generated centrally in bulk and transmitted economically over long distances and is almost pollution free at the consumer level. Electrical energy is generated by conversion of energy available in different forms from different natural sources such as pressure head of water energy, kinetic energy of flowing winds, solar energy, geothermal energy, chemical energy of fuels and nuclear energy of radio-active substances.

The development of power in any country depends upon the available resources in that country. The hydropower totally depends upon the natural sites available and hydrological cycle of the location of country. New sites cannot be humanly created for hydro-power plants. Nuclear power is recently being in developing phase. The development of nuclear power in a country requires advance technological developments and fuel resources. This power source is not much desirable for the developing countries as it is dependent on high technology and they are highly capital based system. The common sources of energy in our country are coal, water, oil and gas. Coal is the most important commercial source of energy in India. Thermal power generation can be most reliable because coal reserves in our country are sufficient to last for some hundreds of years. However the quality of coal produced must be improved.

1.2 Classification of Power plant

Based upon the various factors the power plants can be classified as follows :

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1. According to type of fuel used
   - (i) Steam power plants
   - (ii) Diesel power plants
   - (iii) Gas turbine power plants
   - (iv) Nuclear power plants
   - (v) Hydro-electric power plants
2. According to mobility of plant
   - (i) Stationary power plants
   - (ii) Mobile power plant
3. According to nature of load
   - (i) Base load plants
   - (ii) Peak load plants
4. According to nature of organization
   - (i) Central power plants
   - (ii) Captive power plants

Base load power plants : When capacity factor of plant is greater than 57%, the power plant is known as based load power plant. The capacity factor is the ratio of the average power of generating plant to the rated power over given time interval. This power plant is large capacity plant and working average more than 5000 full power hours working per year. Due to high capacity of this plant have high capital cost but low operating costs.

The thermal (steam) power plants, hydro-electric power plants and nuclear power plants are commonly used as base load power plants.

Peak load power plants : When capacity factor of power plant is less than 23%, the power plant is known as peak load power plant. This plant is small capacity plant and working average is less than 2000 full power hours per year. This plant is operated only to meet the power demands at a time of maximum demand, commonly during extreme summer and extreme winter. This plant requires less investment cost but high operating cost.

The Diesel power plants, gas turbine power plants and pumped storage hydro-electric power plant are commonly used as a peak load plant.

Central power station : These power plants are large capacity plant and generate electric power for general sale to the all consumers. Consumers may be from public, business or industry. The capital cost of these power plants are very high but the operating costs are low. These plants are set up by the State Electricity Boards, Central and State governments, Public sector organizations like National Thermal Power Corporation (NTPC), National Hydro-electric Power Corporation (NHPC), Public-Private sectors (combined).

Captive Power Plants : These type of power plants are run by manufacturing company for its own use and its output is not available for general sale. Captive power plant set by a co-operative society of companies for their own use or for supply of power to the neighboring

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industry.

In the Gujarat, IOC, ONGC, Reliance industries, etc. have their captive power plants.

1.3 Thermal Power Plant

The steam power plants, Diesel engine power plants, Gas turbine power plants and Nuclear power plant are known as Thermal power plant. However, generally steam power plant is called as thermal power plant. The steam power plant can perform two purposes. (i) to generate electricity only (ii) to generate electricity along with production of steam for process heating.

Working of steam power plant :

Steam power plant basically works on the Rankine cycle in which steam and water is working fluid. In the boiler steam is generated from water by using heat of flue gases which is produced by means of burning of coal. The steam which is expanded in a turbine, which produces mechanical power. The output power of turbine is utilized to run the generator. The steam after expansion in turbine is usually condensed in a condenser. The condensed steam (water) is again fed to the boiler and cycle is repeated.

The schematic diagram of **Rankine cycle** is shown in Fig. 1.1. A Rankine cycle is represented on *p-v*, *T-s*, and *h-s* diagram as shown in Fig. 1.2.

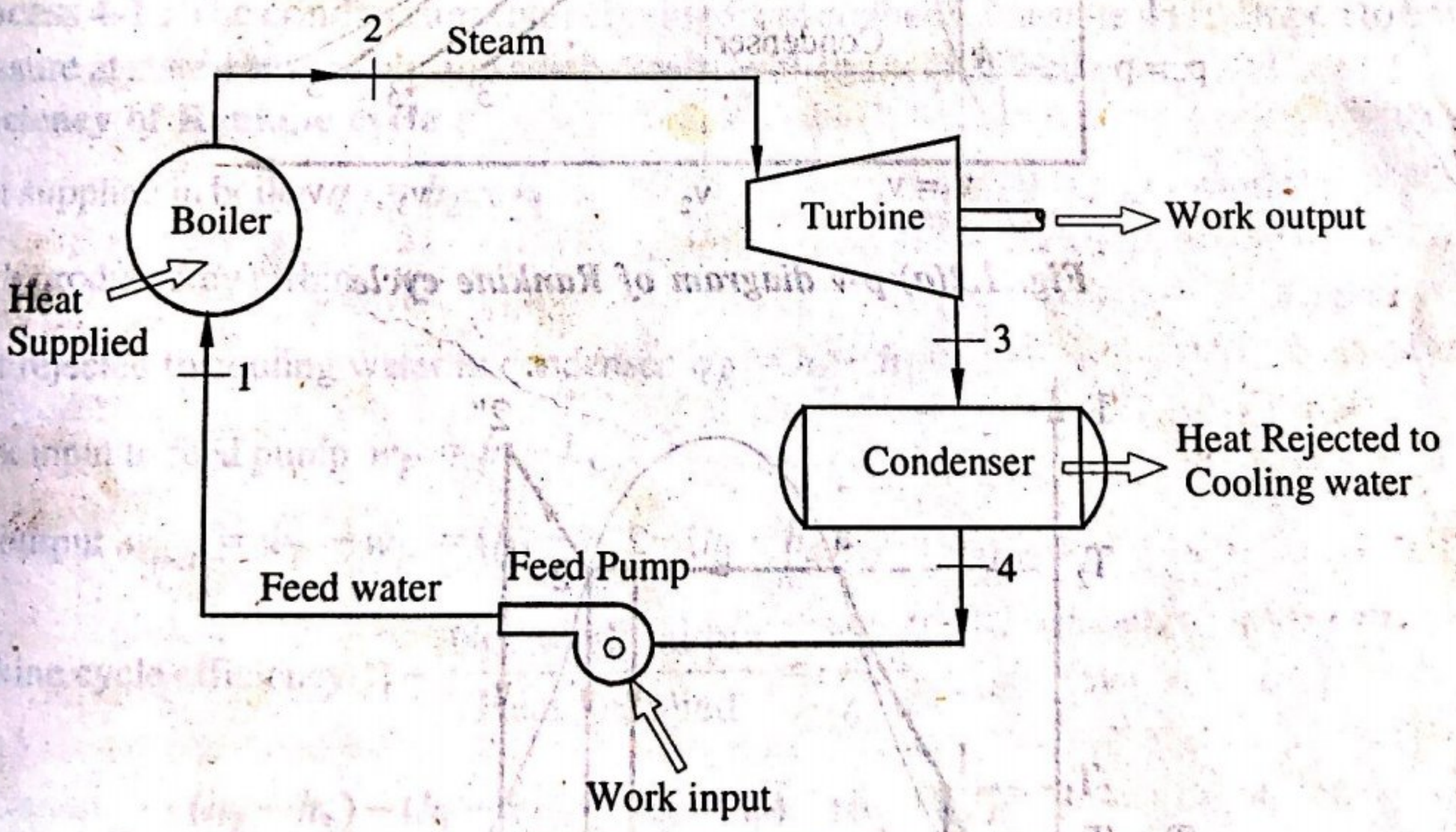


Fig. 1.1 Schematic diagram of Rankine cycle

The main components of cycle are (i) boiler, (ii) turbine, (iii) condenser and (iv) feed pump.

Consider 1 kg of saturated water at pressure p_1 and temperature T_1. The cycle is completed by the following four processes.

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Process 1-2 : The water at constant pressure p_1 is heated in the boiler until the saturated temperature is reached (process 1-a). Saturated water may be converted into dry saturated steam (process a-2), wet steam (a-2′) or superheated steam (process a-2″) at constant pressure p_1.

Process 2-3 : Steam expand reversibly and adiabatically in the turbine from state 2 to state 3 (or 2′ to 3′ or 2″ to 3″ as per condition of steam at outlet of boiler) as shown in Fig. 1.2(a), (b) and (c). During this process pressure and temperature falls from p_2 to p_3 and T_2 to T_3 respectively.

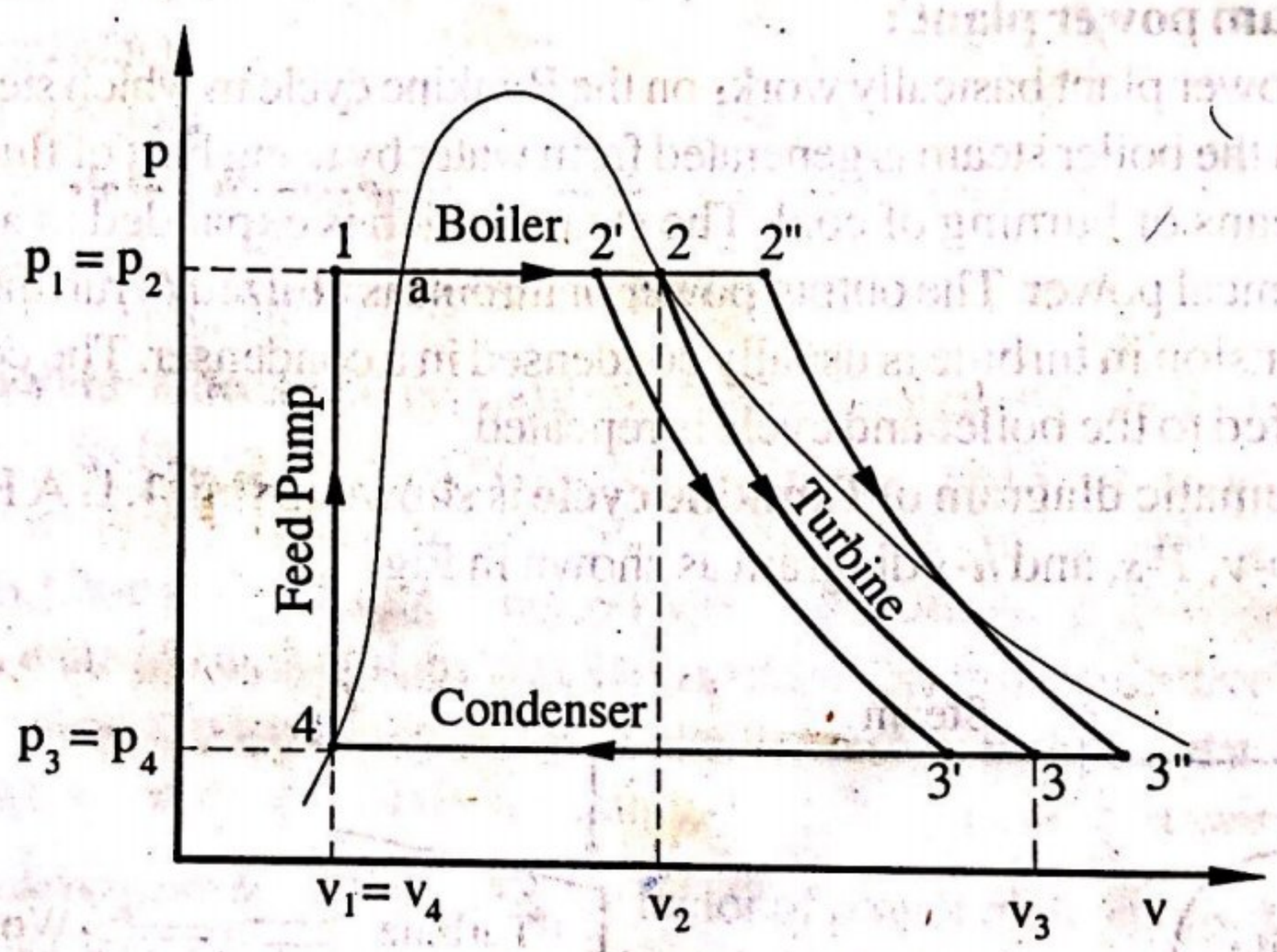


Fig. 1.2(a) p-v diagram of Rankine cycle

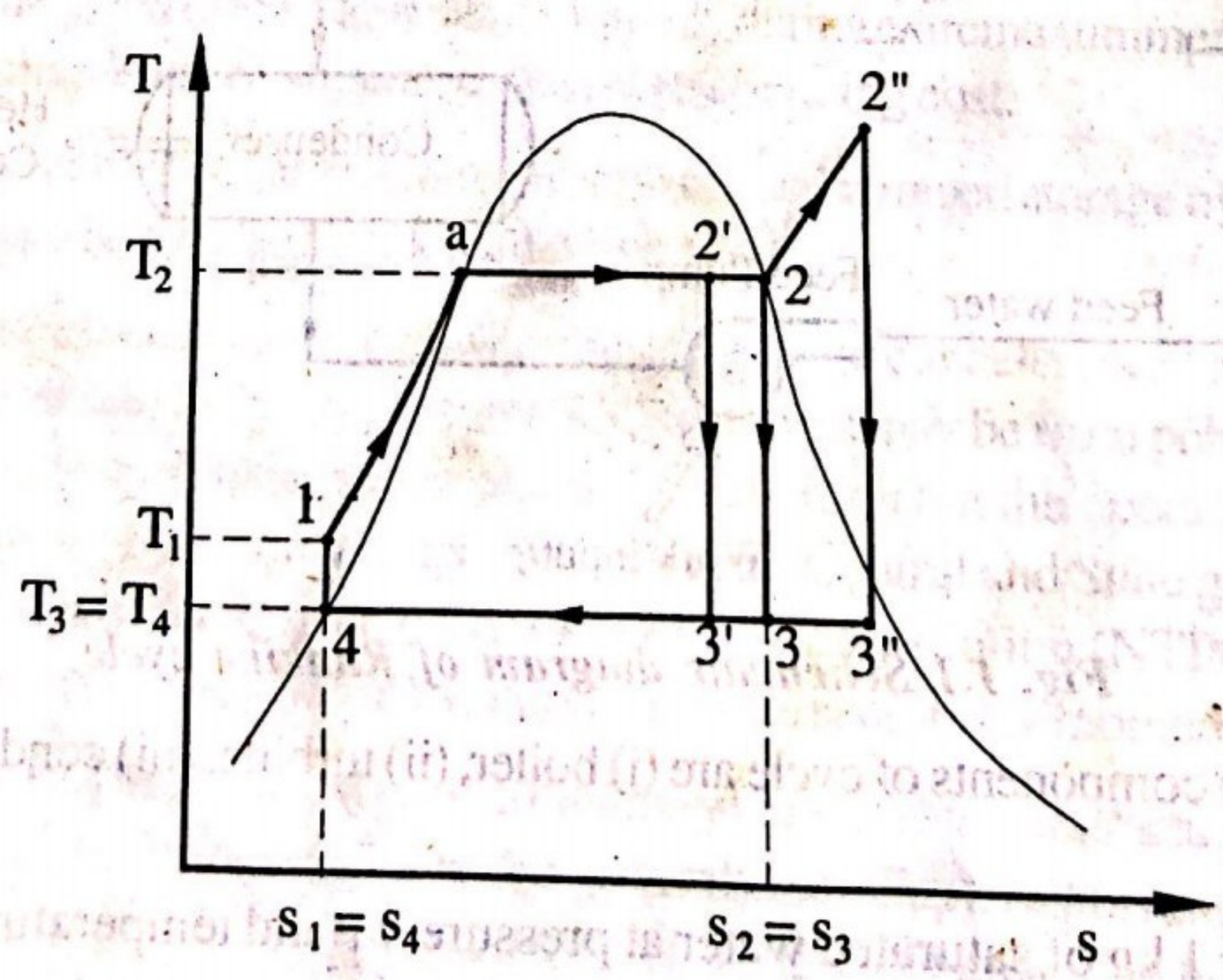


Fig. 1.2 (b) T-s diagram of Rankine cycle

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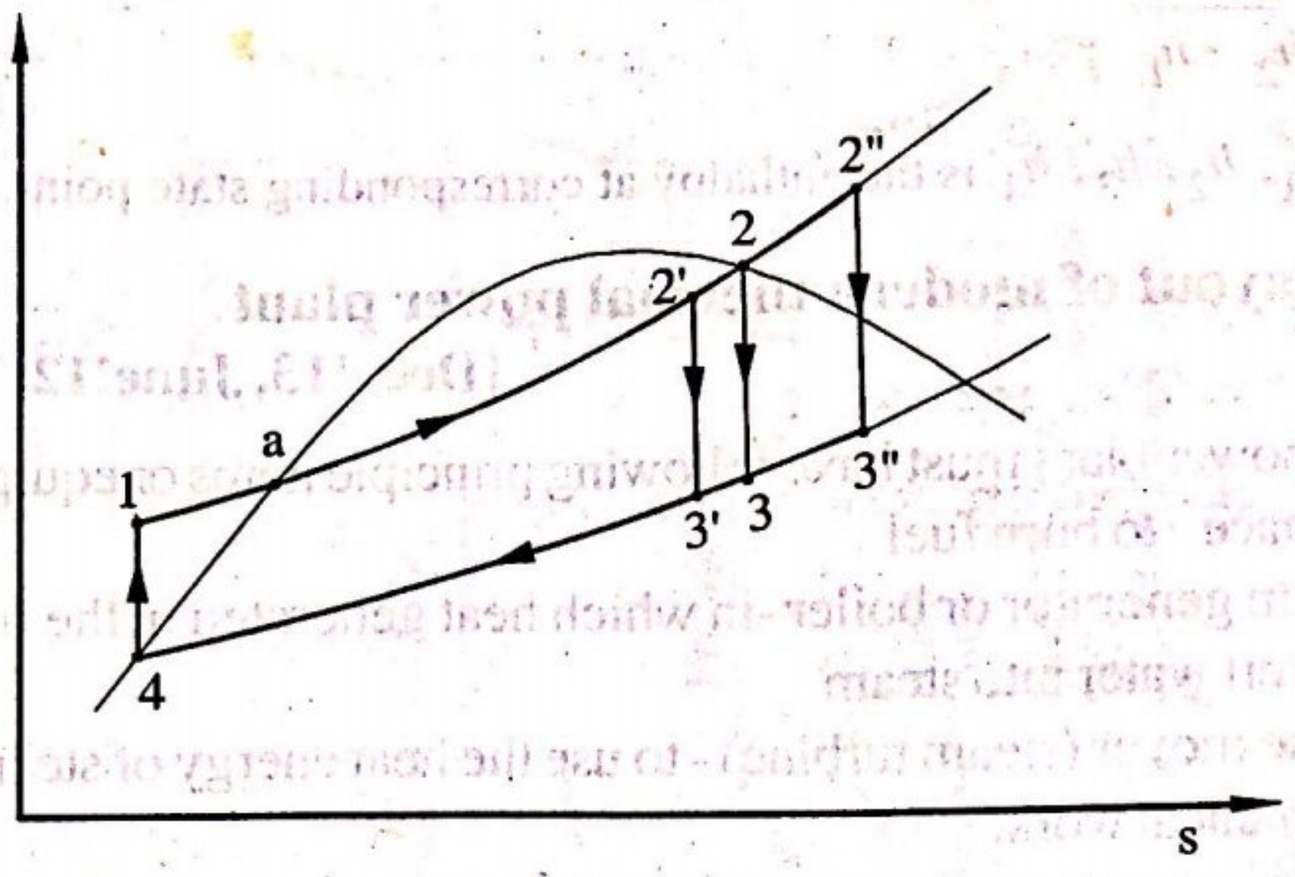


Fig. 1.2 (c) h-s diagram of Rankine cycle

Process 3-4 : The steam coming from turbine at state 3 (or 3′ or 3″) is now isothermally condensed in a condenser and the heat is rejected at constant temperature T_3 and pressure p_3 until the whole steam is condensed into water.

Process 4-1 : The condensate water coming from condenser at state 4 is pumped to boiler pressure at state 1 reversibly and adiabatically with the help of feed pump.

Efficiency of Rankine cycle :

Heat supplied in boiler $q_s = h_2 - h_1$

Work produced by turbine $w_T = h_2 - h_3$

Heat rejected to cooling water in condenser $q_R = h_3 - h_4$

Work input to feed pump $w_P = h_1 - h_4$

Net output $w_{net} = w_T - w_P = (h_2 - h_3) - (h_1 - h_4)$

$$\text{Rankine cycle efficiency } \eta = \frac{\text{Net work output}}{\text{Heat supplied}} = \frac{w_{net}}{q_S}$$

$$\therefore \eta = \frac{(h_2 - h_3) - (h_1 - h_4)}{(h_2 - h_1)} = \frac{(h_2 - h_1) - (h_3 - h_4)}{(h_2 - h_1)}$$

$$\therefore \eta = 1 - \frac{h_3 - h_4}{h_2 - h_1}$$

Usually pump work is very small, hence it may be neglected

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$$\therefore \eta = \frac{h_2 - h_3}{h_2 - h_1}$$

where h_1, h_2, h_3, h_4 is the enthalpy at corresponding state point.

1.4 General layout of modern thermal power plant

[Dec. '13, June'12, Dec.'11, June'11]

A steam power plant must have following principle items or equipments as

(i) Furnace - to burn fuel

(ii) Steam generator or boiler -in which heat generated in the furnace is utilized to convert water into steam

(iii) Prime mover (steam turbine) - to use the heat energy of steam and converts into mechanical work.

(iv) Piping system to convey steam and water.

(v) Electrical generator or alternator - to convert mechanical power into electricity.

Fig. 1.3 shows a schematic arrangement of equipments of a modern coal fired steam power plant. The entire arrangement for the sake of simplicity may be divided into four main circuit as (i) Coal and ash circuit, (ii) Air and flue gases circuit, (iii) Feed water and steam circuit and (iv) Cooling water circuit.

(i) Coal and ash circuit

Coal received in coal storage yard of power station is transferred to the furnace by coal handling plant. After necessary preparation of the coal, from coal handling plant is taken into the boiler bunkers by means of bucket conveyers. Then the coal is stored in the bunkers. Coal from bunkers falls into the hoppers by gravity and finally the requisite quantity of coal either goes on falling directly on grate or where the coal spreaders are provided, coal is spread in the grate up to rear end with help of coal spreaders.

Ash is the product of coal combustion in the furnace, along with flue gases. About 20% of the ash falls in the bottom ash hopper of boiler and is periodically removed mechanically. The ash from hopper is removed to the ash storage by means of scrap conveyors (ash handling plant). Further ash disposed off in accordance with ash disposal arrangement.

(ii) Air and flue gases circuit

The air is taken from the atmosphere by the action of a forced draught fan and passed through an air pre-heater where it is preheated by the flue gases. The preheated air is supplied to the furnace in order to provide air for combustion of fuel. Due to combustion of fuel (coal) hot gases (flue gases) are produced. The flue gases from the furnace pass over boiler tubes and superheater tubes. Then flue gases pass though economiser (in which water is heated) and air preheater (in which air is heated). Flue gases is then passed through dust and fly ash collecting device where suspended dust and fly ash are removed from it. The fly ash is separated from flue gases by means of ash precipitator. Finally, flue gases is exhausted to the atmosphere through chimney.

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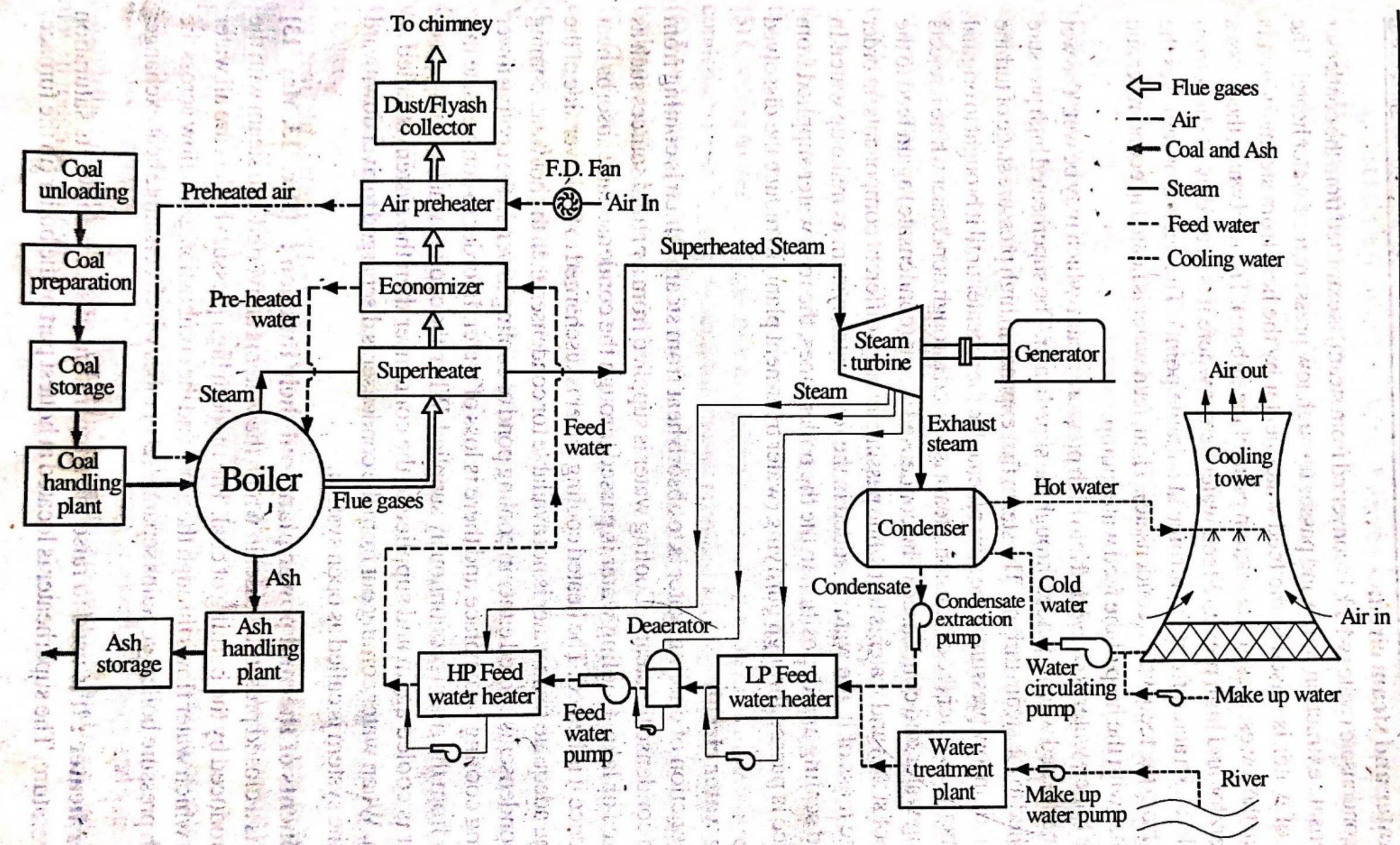


Fig. 1.3 General layout of a steam power plant [May '13]

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(iii) Feed water and steam circuit

The condensate (water which is converted from steam) is extracted from the condenser by the condensate extraction pump and passes through low pressure feed water heater. The temperature of water is raised in the feed water heater by the heat from steam (which is extracted from the lowest pressure extraction point of the turbine). The feed water, then passes through deaerator and high pressure feed water heater where it is heated by the heat from steam (which is extracted from the turbine at suitable point). The deaerator removes the dissolved O_2 from the feed water (condensate). Then, feed water is heated in economiser and fed to the boiler with the help of the feed water pump.

In the boiler, water is converted into high pressure steam which may be wet. The wet steam gets superheated when it passes through the superheater. The superheated high pressure steam is supplied to the steam turbine. The steam first expands in high pressure stage of turbine and then expands in low pressure stage after reheated in the reheater until it brought to original dryness or superheat. After the expansion, steam is passed through condenser where it rejects heat to cooling water and condensed. The condensed steam (condensate) again feed to the boiler. A part of steam and water is lost while passing through different components, in order to compensate mass of it, the water is added in the feed water system as make up water. In order to avoid scaling and deposition in the the boiler tubes, the feed water supplied from external source is passed through purifying (water treatment plant) to reduce the dissolved salts.

(iv) Cooling water circuit

The function of cooling water is to absorb heat from steam which is coming from turbine and to condense the steam. Cooling water is supplied from a natural sources such as river, canal, sea or lake. Cooling water is passed through the condenser and absorbs heat from steam hence it is heated. The heated cooling water is discharged to the lower side of the river. When the adequate fresh water is not available for condenser, then the hot water coming out from the condenser is cooled either in cooling pond or cooling tower. In the cooling tower the evaporating cooling takes place and there is loss of water. In order to compensate this loss, the water from the river is continuously supplied. When the cooling water coming out of the condenser is cooled again and supplied to the condenser then the system is called as closed system. When water coming out from the condenser is discharged to river downward side directly, the system is called as open system.

Main components of thermal power plant and their function : **[Dec. '13, May '13]**

(1) Boiler : It is a closed pressure vessel in which the water is converted into steam with the help of heat produced by the burning of coal. The modern high pressure boilers are all water tube boilers in which water flows through the tubes and the combustion gases flow across the tubes. The high pressure boilers contains separate set of tubes which constitute heat exchanger as follows :

(a) Super heater : It is used to raise the temperature of steam above its saturation temperature. The superheater is located at hottest part of the boiler (in the furnace)

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so that heat is transferred to steam by superheater from the hot gases.

(b) Reheater : It is used to raise the temperature of steam after it has expanded in the high pressure turbine. After reheating the steam supplied to the low pressure steam turbine.

(c) Economizer : The function of economizer is to increase temperature of feed water using waste heat of flue gases leaving the boiler through chimney. It is located before airpreheater and after superheater and reheater in the path of the flue gases.

(d) Air preheater : The function of air preheater is to increase the temperature of air before it supply to the furnace using heat from flue gases passing through chimney.

(2) Steam turbine : The function of steam turbine is to convert the heat energy of steam into rotational power at turbine shaft. In modern thermal power plant steam turbine generally has a three cylinder tendum compound machine consisting high pressure stage cylinder (HP), intermediate pressure stage (I.P.) and low pressure stage cylinder (L.P.).

(3) Generator : The function of electric generator is to convert mechanical (rotational) power of turbine shaft into electric energy. The generator shaft is coupled to the turbine shaft.

(4) Condenser : A condenser is a heat exchanger, consists of large cylindrical vessel containing large number of brass tubes through which cooling water circulated and steam flowing outside the surface of tubes. The function of condenser is to condense the steam which coming form the low pressure turbine.

(5) Cooling tower : The function of cooling tower is to cool the hot water coming from the condenser. In the cooling tower the hot water is cooled with the help of the atmospheric air.

(6) Feed water pump : It is used to supply feed water (coming from the condenser) to the boiler at a desired high pressure.

(7) Circulating water pump : It is used to circulate cooling water in the condenser and cooling tower closed circuit.

(8) Coal handing plant

Wagon trippler : It is used to unload the coal when coal transported to the plant site by rail wagon.

Crusher house : It is used to crush coal to a size of 10 mm. Then coal is supplied to boiler raw coal bunkers.

Coal mill : It is used to pulverise the raw coal into a fine powder before it is burnt in the boiler furnace.

(9) Draught system : The function of draught system is to create pressure difference which is required to introduce desired quantity of air to the boiler furnace and to carry away the burned product of combustion and discharge to atmosphere.

Forced fraught (F.D) fan : F.D. fan is placed in front of boiler and it is used to push the air through the air preheater and then the furnace of boiler.

Induced draught (I.D.) fan : I. D. fan is placed near the base of chimney and it draws flue gases through the interior of the boiler and dust extracting equipment and forces the flue gases to move through the chimney.

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(10) Ash precipitators : The function of ash precipitator is to remove flyash and dust particles from the flue gases before these escape into the atmosphere through chimney to avoid air pollution.

(11) Chimney : It is ferro-concrete structure lined with fire bricks. It is used to escape flue gases to atmosphere at 30 m above the ground.

(12) Water treatment plant : The function of water treatment plant is to remove impurities of water before water supply to the boiler.

(13) Switch yard : The switch yard houses transformers, circuit breakers and switches for connecting and disconnecting the transformer and circuit breakers. The electricity generated by the 11 kV generator is stepped up by unit transformers to 132/220 kV for the further transmission through high tension lines and fed to the regional grid.

(14) Control room : The control room houses all the necessary measuring instruments for each panel of generator (alternator) and feeder, synchronizing gear, protective gear, automatic voltage regulator, communication arrangement etc. The performance of all the plant equipments is constantly monitored here with the help of sophisticated instrumentation and controllers.

1.5 Site selection of thermal power plant [June '11]

Following factors must be considered while selecting the site for a thermal power plant.

(1) Availability of coal : Modern coal based steam power plant requires huge quantity of coal per year. A rough idea, a thermal power plant of 100 MW capacity requires 1200 to 1500 tones of coal per day. Hence a plant consuming such a large quantity of coal should be located as far as possible near the coal mines in order to save the coal transportation costs. Also, if plant is located away from coal mines it may create problems due to failure of transportation system or strike at mines etc. For these reasons considerable amount of coal (atleast 15 day coal supply) must be stored at the power station. However it increases space requirements, staffs, costs and losses of coal storage. In case it is not possible to locate the plant near the coal mines, then the plant should be connected to the railway.

(2) Availability of water : In the steam power plant, water and steam is working fluid, hence large quantity of feed or make up water is required. Also huge amount of water required for condenser (cooling water) and for disposal of ash.

A rough idea, a thermal power plant of 100 MW capacity, the 30,000 to 50,000 tones cooling water required per hour. If the cooling tower is used than also 800 to 1000 tones cooling water required per hour. The make up water required to compensate the loss of feed water is also 10 to 15 tones per hour. Further the water must be pure as possible to reduce cost of water treatment for feed water. Therefore, while selecting site for the steam power plant, above factor must be considered carefully. It is therefore necessary to locate the plant near the water source which will be able to supply required quantity of water throughout the year.

(3) Ash disposal facilities : Ash is the main waste product of steam power plant. The ash

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handling problem is more serious than coal handling, particularly in India (coal available in India is having large percentage of ash as 20 to 40%) because ash comes out in hot condition and it is highly corrosive. It's effect on atmospheric pollution are more serious as human health is concerned. A 100 MW steam power plant, produces approximately 350 to 500 tones ash per day, it requires 2.5 hectares area per year if the ash is dumped to a height of 6.5 meters. If the site as near the coal mines, it can be dumped into the disused mines. In case of site located near the river, sea or lakes ash can be dumped into it. Presently ash from power station is used for many industrial processes and brick making for construction work.

(4) Land requirement : Land required not only for setting up the plant but for other purposes such as staff colony, coal storage, ash disposal etc. For 100 MW plant, the land requirement may be 300 to 500 acres. As the cost of land adds up to the final cost of the plant, it should be available at a reasonable price. Land should be also available for future extension.

(5) Nature of land : The type of land to be selected should have good bearing capacity as it has to withstand not only the dead load of the plant but also the forces transmitted to the foundations due to operation of heavy machinery of plant. The minimum bearing capacity of the land should be 1×10^6 N/m^2.

(6) Load centre : The site of power plant should be as near as possible to the loads so that the transmission cost and losses are minimum. However, a plant can not be located near all the loads. In this case C.G. of the loads is determined with reference two arbitrarily chosen axes, this C.G. is known as load centre. Hence in order to minimise the cost and losses the plant should be located as far as possible near the load centre.

(7) Transport facilities : The power plant should be near the railway line for bringing the heavy machinery for installation and for bringing the coal.

(8) Availability of labour : At the time of construction of plant enough labour is requires. Hence, there should be skilled and unskilled labours available at reasonable rates near the site of the plant.

(9) Size of the plant : In case of small power plant, cost of transmission line is relatively high, hence it should be near to the load centre. But, in case of large power plant the cost of transportation of coal, cost of water are relatively high compared to the cost of transmission line. Hence power plant should be near the coal mine as well as water source.

(10) Public problems : Site selected should not any short of nuisance to the neighbouring areas due to emission of smoke, noise, water vapour etc. Hence plant should be located far away from the towns.

(11) Future extensions : The choice of the site should allow for economical extensions consistence with the estimated growth of load.

A site selected may not necessarily satisfy all the factors mentioned above but however a suitable compromise must be made between the major factors.

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1.6 Present status of power generation in India

1.6.1 Energy sources and present power installed capacity in India

India is fairly rich in natural sources like coal, lignite, natural gas and has immense water power resources. India's total mean annual river flows are about 1675×10^9 m^3 of which the usable resources are 555×10^9 m^3. Out of this 60% contribution from Himalayan rivers (Ganga, Indus and Brahmaputra), 16% from central rivers (Narmada, Tapti and Mahanadi) and remaining from the rivers draining the Deccum plateau (Godavari, Krishna and Cauvery).

Some oil reserves have already been discovered and intense exploration is being undertaken in various states. India has vast reserves of natural gas estimated at over one hundred billion m^3. Natural gas is produced in Assam, Triputa, Gujarat and Western offshore. Though India is deficient in uranium (fuel of Nuclear power plant), but has rich deposits of thorium which can be utilized in future. Since independence the country has made tremendous progress in the development of electric power and today India has the largest power system among the developing countries.

Northern region of India :

In this region, Jammu and Kashmir, Punjab, Uttaranchal are very rich in water power resources and therefore power development should be based on this source. Haryana and Himachal Pradesh has a sizeable potential for water power. Expansion of base load requirements in Punjab, Haryana and Himachal Pradesh should be met by an nuclear power plant located in this area in preference to a conventional steam power plant which will not be economical. This area can receive backing from Delhi and Badarpur steam power plants. In Jammu and Kashmir conventional steam power plant and Nuclear power plant will be totally unjustified. Uttar Pradesh has sufficient water power resources and therefore this should be main source for power generation. The water power can be supplemented by conventional steam power plants which should be preferably located in Eastern part of the U.P.. Rajasthan is deficient both in water potential and coal resources. Therefore the power programme should be based on nuclear power plant only. The present power installed capacity (MW) as on 31-01-2010 in Northern region is given in Table 1.1 and graphical representation given in Fig. 1.4.

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Table 1.1
Installed capacity (MW) of power in Northern region of India

State	Ownership Sector	Thermal Power Plant			Hydro Power	Nuclear Power	Renewable Energy	Total
		Coal	Gas	Diesel				
Delhi	State	320.00	600.40	0.00	0.00	0.00	0.00	920.40
	Central	2467.96	207.61	0.00	581.62	47.08	0.00	3304.27
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**2787.96**	**808.01**	**0.00**	**581.62**	**47.08**	**0.00**	**4224.67**
Haryana	State	2132.50	0.00	3.92	884.51	0.00	68.70	3089.63
	Central	402.99	535.29	0.00	443.17	76.16	0.00	1457.61
	Private	0.00	0.00	0.00	0.00	0.00	6.00	6.00
	Sub. Total	**2535.49**	**535.29**	**3.92**	**1327.68**	**76.16**	**74.70**	**4553.24**
Himachal	State	0.00	0.00	0.13	393.60	0.00	255.42	649.15
	Central	118.30	61.88	0.00	760.34	14.08	0.00	954.60
	Private	0.00	0.00	0.00	386.00	0.00	0.00	386.00
	Sub. Total	**118.30**	**61.88**	**0.13**	**1539.94**	**14.08**	**255.42**	**1989.75**
Jammu & Kashmir	State	0.00	175.00	8.94	780.00	0.00	129.33	1093.27
	Central	263.70	129.14	0.00	700.53	68.00	0.00	1161.37
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**263.70**	**304.14**	**8.94**	**1480.53**	**68.00**	**129.33**	**2254.64**
Panjab	State	2630.00	0.00	0.00	2230.23	0.00	217.95	5078.18
	Central	578.19	263.92	0.00	732.66	151.04	0.00	1725.81
	Private	0.00	0.00	0.00	0.00	0.00	58.25	58.25
	Sub. Total	**3208.19**	**263.92**	**0.00**	**2962.89**	**151.04**	**276.20**	**6862.24**
Rajasthan	State	3240.00	443.80	0.00	987.96	0.00	30.25	4702.01
	Central	649.48	221.23	0.00	466.84	469.00	0.00	1806.55
	Private	135.00	0.00	0.00	0.00	0.00	880.40	1015.40
	Sub. Total	**4024.48**	**665.03**	**0.00**	**1454.80**	**469.00**	**910.65**	**7523.96**
Uttar Predesh	State	4072.00	0.00	0.00	524.10	0.00	25.10	4621.20
	Central	2540.84	549.97	0.00	1073.32	203.72	0.00	4367.85
	Private	0.00	0.00	0.00	0.00	0.00	545.50	545.50
	Sub. Total	**6612.84**	**549.97**	**0.00**	**1597.42**	**203.72**	**570.60**	**9534.55**
Uttran-chal	State	0.00	0.00	0.00	1252.15	0.00	132.92	1385.07
	Central	261.26	69.35	0.00	267.03	16.28	0.00	613.92
	Private	0.00	0.00	0.00	400.00	0.00	0.00	400.00
	Sub. Total	**261.26**	**69.35**	**0.00**	**1919.18**	**16.28**	**132.92**	**2398.99**
Chandi-garh	State	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Central	27.09	15.32	0.00	46.74	4.84	0.00	93.99
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**27.09**	**15.32**	**0.00**	**46.74**	**4.84**	**0.00**	**93.99**

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Western region of India :

Gujarat is deficient both in water power and coal resources, therefore, the nuclear power is to form the base. The small water potential and natural gas available in the state can serve the peaking capacity. Madhya Pradesh has sufficient coal reserves and water power resources. Therefore, the power programme should be based on combination of these resources. Maharashtra has sizeable water potential. Since, in Maharashtra, coal resources are laking, the backing power has to be provided by nuclear power. The present power installed capacity (MW) as on 31-01-2010 in Western region is given in Table 1.2 and graphical representation given in Fig. 1.5. Gujarat state is rich in gas resources. Therefore, gas based power has a good contribution in total generation as shown in Fig. 1.6.

Table-1.2: Installed capacity (MW) of power in Western region of India

State	Ownership Sector	Thermal Power Plant			Hydro Power	Nuclear Power	Renewable Energy	Total
		Coal	Gas	Diesel				
Goa	State	0.00	0.00	0.00	0.00	0.00	0.05	0.05
	Central	277.03	0.00	0.00	0.00	25.80	0.00	302.83
	Private	0.00	48.00	0.00	0.00	0.00	30.00	78.00
	Sub. Total	**277.03**	**48.00**	**0.00**	**0.00**	**25.80**	**30.05**	**380.88**
Daman & Div	State	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Central	19.04	4.20	0.00	0.00	7.38	0.00	30.62
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**19.04**	**4.20**	**0.00**	**0.00**	**7.38**	**0.00**	**30.62**
Gujarat	State	4190.00	892.72	17.28	772.00	0.00	24.30	5896.30
	Central	1508.89	424.27	0.00	0.00	559.32	0.00	2492.48
	Private	980.00	2577.50	0.20	0.00	0.00	1619.61	5177.31
	Sub. Total	**6678.89**	**3894.49**	**17.48**	**772.00**	**559.32**	**1643.91**	**13566.09**
Madhya Pradesh	State	2807.50	0.00	0.00	1703.66	0.00	71.76	4582.92
	Central	1474.60	257.18	0.00	1520.00	273.24	0.00	3525.02
	Private	0.00	0.00	0.00	0.00	0.00	215.90	215.90
	Sub. Total	**4282.10**	**257.18**	**0.00**	**3223.66**	**273.24**	**287.66**	**8323.84**
Chhatis-garth	State	2060.00	0.00	0.00	120.00	0.00	19.05	2199.05
	Central	723.00	0.00	0.00	0.00	47.52	0.00	770.52
	Private	1300.00	0.00	0.00	0.00	0.00	187.90	1487.90
	Sub. Total	**4083.00**	**0.00**	**0.00**	**120.00**	**47.52**	**206.95**	**4457.47**
Maharastra	State	6800.00	912.00	0.00	2884.84	0.00	223.72	10820.56
	Central	2003.05	2623.93	0.00	0.00	690.14	0.00	5317.12
	Private	1900.00	180.00	0.00	447.00	0.00	2204.25	4731.25
	Sub. Total	**10703.05**	**3715.93**	**0.00**	**3331.84**	**690.14**	**2427.97**	**20868.93**
Dadra & Nagar Haveli	State	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Central	22.04	27.10	0.00	0.00	8.46	0.00	57.60
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**22.04**	**27.10**	**0.00**	**0.00**	**8.46**	**0.00**	**57.60**

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Southern region of India :

Karnataka, Kerala, Andhra Pradesh are very rich in water power resources and therefore power development programmes should be based on this source. Tamil Nadu is deficient in coal resources and it has already exhausted its limited water power potential. Therefore, the future power programme should be based largely on Nuclear power plants. The present power installed capacity (MW) as on 31-01-2010 in Sourthern region is given in Table 1.3 and graphical representation given in Fig. 1.7.

Table-1.3: Installed capacity (MW) of power in Southern region of India

State	Ownership Sector	Thermal Power Plant			Hydro Power	Nuclear Power	Renewable Energy	Total
		Coal	Gas	Diesel				
Andhra Pradesh	State	3882.50	0.00	0.00	3617.53	0.00	188.43	7688.46
	Central	2377.38	0.00	0.00	0.00	214.28	0.00	2591.66
	Private	0.00	2580.40	36.80	0.00	0.00	511.58	3128.78
	Sub. Total	**6259.88**	**2580.40**	**36.80**	**3617.53**	**214.28**	**700.01**	**13408.90**
Karnataka	State	1970.00	0.00	127.92	3599.80	0.00	527.15	6224.87
	Central	1072.67	0.00	0.00	0.00	195.36	0.00	1268.03
	Private	860.00	220.00	106.50	0.00	0.00	1706.54	2893.04
	Sub. Total	**3902.67**	**220.00**	**234.42**	**3599.80**	**195.36**	**2233.69**	**10385.94**
Kerala	State	0.00	0.00	234.60	1781.50	0.00	138.76	2154.86
	Central	765.38	359.58	0.00	0.00	78.10	0.00	1203.06
	Private	0.00	174.00	21.84	0.00	0.00	0.00	195.84
	Sub. Total	**765.38**	**533.58**	**256.44**	**1781.50**	**78.10**	**138.76**	**3553.76**
Tamil Nadu	State	2970.00	523.20	0.00	2108.20	0.00	85.55	5686.95
	Central	2299.81	0.00	0.00	0.00	478.50	0.00	2778.31
	Private	250.00	503.10	411.66	0.00	0.00	4778.56	5943.32
	Sub. Total	**5519.81**	**1026.30**	**411.66**	**2108.20**	**478.50**	**4864.11**	**14408.58**
NLC	State	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Central	100.17	0.00	0.00	0.00	0.00	0.00	100.17
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**100.17**	**0.00**	**0.00**	**0.00**	**0.00**	**0.00**	**100.17**
Pondi-chery	State	0.00	32.50	0.00	0.00	0.00	0.00	32.50
	Central	207.01	0.00	0.00	0.00	16.28	0.00	223.29
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**207.01**	**32.50**	**0.00**	**0.00**	**16.28**	**0.00**	**255.79**

Eastern region of India :

Bihar, Jharkhand and West Bengal has abundant coal reserves. Therefore, the major power development programme should be based on conventional steam power plant. In Bihar a small amount of water power is also available which can be provide the peak loads. West

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Bengal is deficient in water potential. In Orissa, both water power and coal resources are available and therefore power development programme should be based on a combination of these sources. The present power installed capacity (MW) as on 31-01-2010 in Eastern region is given in Table 1.4 and graphical representation given in Fig. 1.8.

Table-1.4: Installed capacity (MW) of power in Eastern region of India

State	Ownership Sector	Thermal Power Plant			Hydro Power	Nuclear Power	Renewable Energy	Total
		Coal	Gas	Diesel				
Bihar	State	530.00	0.00	0.00	0.00	0.00	54.60	584.60
	Central	1131.70	0.00	0.00	129.43	0.00	0.00	1261.13
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**1661.70**	**0.00**	**0.00**	**129.43**	**0.00**	**54.60**	**1845.73**
Jharkhand	State	1190.00	0.00	0.00	130.00	0.00	4.05	1324.05
	Central	187.88	0.00	0.00	70.93	0.00	0.00	258.81
	Private	360.00	0.00	0.00	0.00	0.00	0.00	360.00
	Sub. Total	**1737.88**	**0.00**	**0.00**	**200.93**	**0.00**	**4.05**	**1942.86**
West Bengal	State	4780.00	100.00	12.06	977.00	0.00	144.50	6013.56
	Central	634.96	0.00	0.00	139.30	0.00	0.00	774.26
	Private	1341.38	0.00	0.14	0.00	0.00	20.35	1361.87
	Sub. Total	**6756.34**	**100.00**	**12.20**	**1116.30**	**0.00**	**164.85**	**8149.69**
DVC	State	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Central	3313.10	90.00	0.00	193.26	0.00	0.00	3596.36
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**3313.10**	**90.00**	**0.00**	**193.26**	**0.00**	**0.00**	**3596.36**
Orissa	State	420.00	0.00	0.00	2083.92	0.00	64.30	2568.22
	Central	1408.10	0.00	0.00	105.01	0.00	0.00	1513.11
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**1828.10**	**0.00**	**0.00**	**2188.93**	**0.00**	**64.30**	**4081.33**
Sikkim	State	0.00	0.00	5.00	0.00	0.00	47.11	52.11
	Central	68.10	0.00	0.00	75.27	0.00	0.00	143.37
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**68.10**	**0.00**	**5.00**	**75.27**	**0.00**	**47.11**	**195.48**

North-Eastern region of India :

Assam and Meghalaya are very rich in water power resources, therefore the power development should be based on this sources. Assam and Tripura are also having very good availability of natural gas therefore the power can be developed by gas turbines using natural gas. The Manipur, Nagaland and Mizoram are laking in coal, natural gas and water power, therefore these power developed using diesel power plant. The present power installed capacity

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(MW) as on 31-01-2010 in the North-Eastern region is given in Table 1.5 and graphical representation given in Fig. 1.9.

Table-1.5: Installed capacity (MW) of power in North-Eastern region of India

State	Ownership Sector	Thermal Power Plant			Hydro Power	Nuclear Power	Renewable Energy	Total
		Coal	Gas	Diesel				
Assam	State	60.00	239.00	20.69	100.00	0.00	27.11	446.80
	Central	0.00	177.82	0.00	329.72	0.00	0.00	507.54
	Private	0.00	24.50	0.00	0.00	0.00	0.00	24.50
	Sub. Total	**60.00**	**441.32**	**20.69**	**429.72**	**0.00**	**27.11**	**978.84**
Arunachal Predesh	State	0.00	0.00	15.88	0.00	0.00	67.42	83.30
	Central	0.00	21.05	0.00	97.57	0.00	0.00	118.62
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**0.00**	**21.05**	**15.88**	**97.57**	**0.00**	**67.42**	**201.92**
Meghalaya	State	0.00	0.00	2.05	156.00	0.00	31.03	189.08
	Central	0.00	25.96	0.00	74.58	0.00	0.00	100.54
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**0.00**	**25.96**	**2.05**	**230.58**	**0.00**	**31.03**	**289.62**
Tripura	State	0.00	127.50	4.85	0.00	0.00	16.01	148.36
	Central	0.00	33.34	0.00	62.37	0.00	0.00	95.71
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**0.00**	**160.84**	**4.85**	**62.37**	**0.00**	**16.01**	**244.07**
Manipur	State	0.00	0.00	45.41	0.00	0.00	5.45	50.86
	Central	0.00	25.96	0.00	80.98	0.00	0.00	106.94
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**0.00**	**25.96**	**45.41**	**80.98**	**0.00**	**5.45**	**157.80**
Nagaland	State	0.00	0.00	2.00	0.00	0.00	28.67	30.67
	Central	0.00	19.19	0.00	53.32	0.00	0.00	72.51
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**0.00**	**19.19**	**2.00**	**53.32**	**0.00**	**28.67**	**103.18**
Mizoram	State	0.00	0.00	51.86	0.00	0.00	28.47	80.33
	Central	0.00	16.28	0.00	34.31	0.00	0.00	50.59
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**0.00**	**16.28**	**51.86**	**34.31**	**0.00**	**28.47**	**130.92**

Islands of India :

In Andaman & Nicobar, the power is generated by diesel power plant and some of renewable energy sources. Lakshdweep fulfils their requirement only by diesel power plant. Islands power installed capacity (MW) as on 31-01-2010 is given in Table 1.6.

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Table-1.6: Installed capacity (MW) of power in Islands of India

State	Ownership Sector	Thermal Power Plant			Hydro Power	Nuclear Power	Renewable Energy	Total
		Coal	Gas	Diesel				
Andaman & Nicobar	State	0.00	0.00	40.05	0.00	0.00	5.25	45.30
	Central	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Private	0.00	0.00	20.00	0.00	0.00	0.00	20.00
	Sub. Total	**0.00**	**0.00**	**60.05**	**0.00**	**0.00**	**5.25**	**65.30**
Laksha-dweep	State	0.00	0.00	9.97	0.00	0.00	0.00	9.97
	Central	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Private	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Sub. Total	**0.00**	**0.00**	**9.97**	**0.00**	**0.00**	**0.00**	**9.97**

All India installed capacity is given in Table 1.7.

Table-1.7: Installed capacity (MW) of power of all India

	Ownership Sector	Thermal Power Plant			Hydro Power	Nuclear Power	Renewable Energy	Total
		Coal	Gas	Diesel				
All India	State	44054.5	4046.12	602.61	27087	0	2662.41	78452.64
	Central	26903.72	6159.57	0	8038.3	3644.58	0	44746.17
	Central Un-allocated	4011.28	542.66	0	527.1	475.42	0	5556.46
	Private	7126.38	6307.5	597.14	1233	0	12764.84	28028.86
	Sub. Total	82095.88	17055.85	1199.75	36885.4	4120	15427.25	156784.1

India has a total installed capacity of power is 1,56,784 MW (as on 31-01-2010), out of these 64% of thermal, 23% hydro, 3% Nuclear and 10% of RES as shown in Fig. 1.10 and Fig. 1.11. Fig. 1.12 represents the installed capacity of power of state government, central and private sectors of India. The state wise central allocation of power is given in Fig. 1.13.

From above data it is cleared that the two major sources of electric power in India are coal (thermal) and water. As a national policy greatest emphasis should be laid on the development of hydro resources because of operating cost of hydro power plant is very less. However high investment cost and hydro power plant takes about 8 to 10 years for it completion. Hence, thermal power plant is the reasonable option to hydro power plant due to availability of fuel in India. Also it is very difficult to maintain a uniform rate of power production from hydro power plant due to wide fluctuation in annual rail fall and run off.

Installed capacity (MW)

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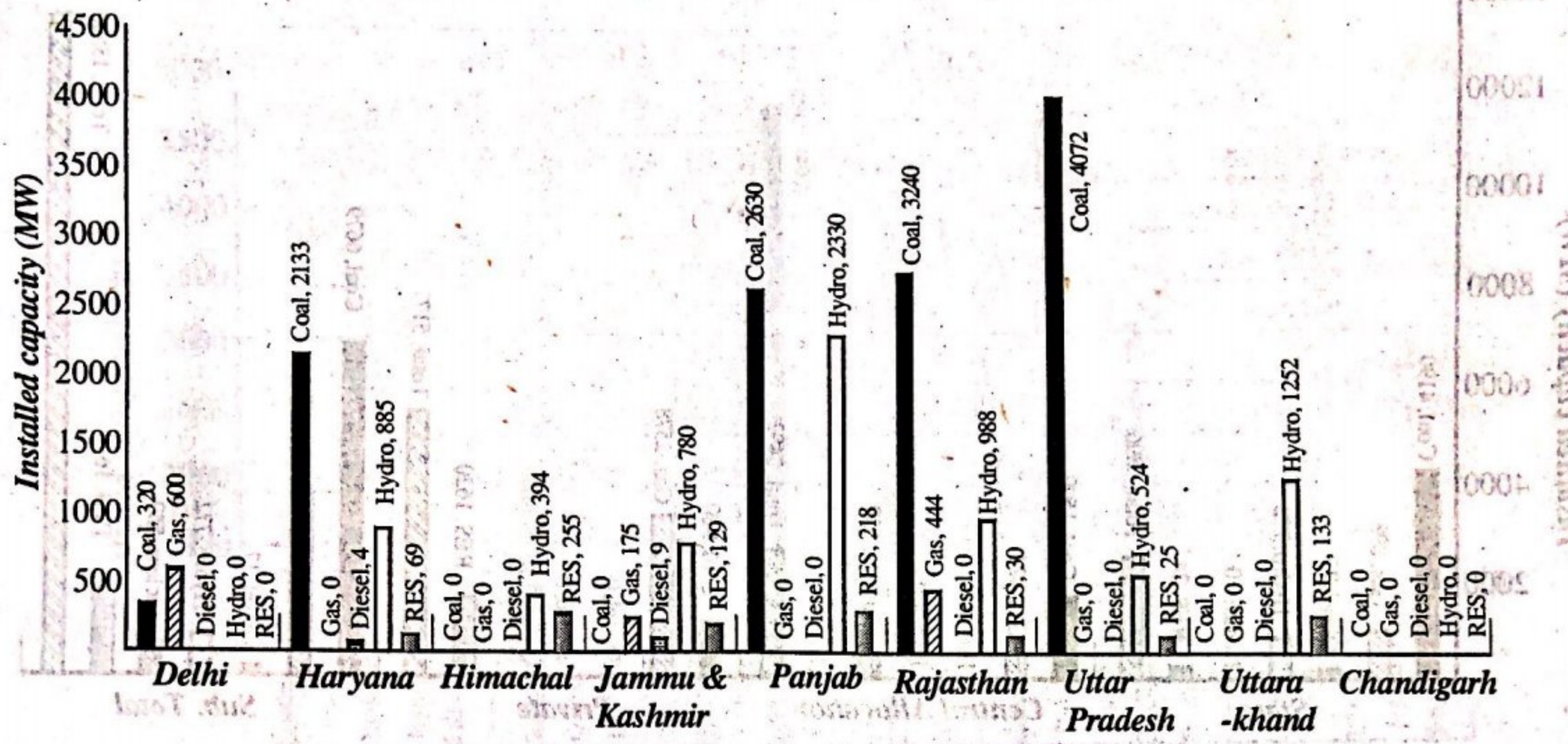


Fig. 1.4 Installed capacity (MW) of power in Northern region of India

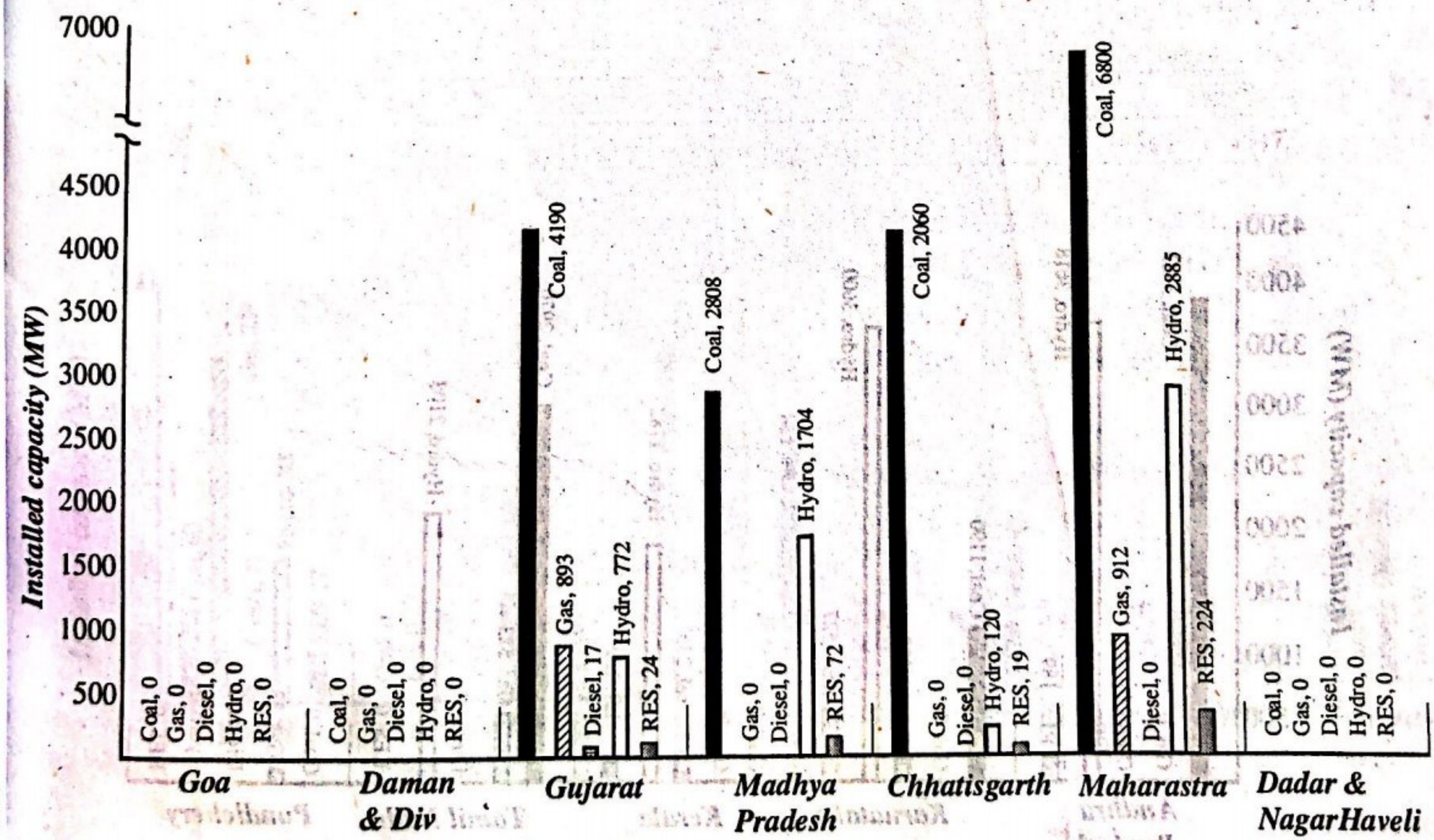


Fig. 1.5 Installed capacity (MW) of power in Western region of India

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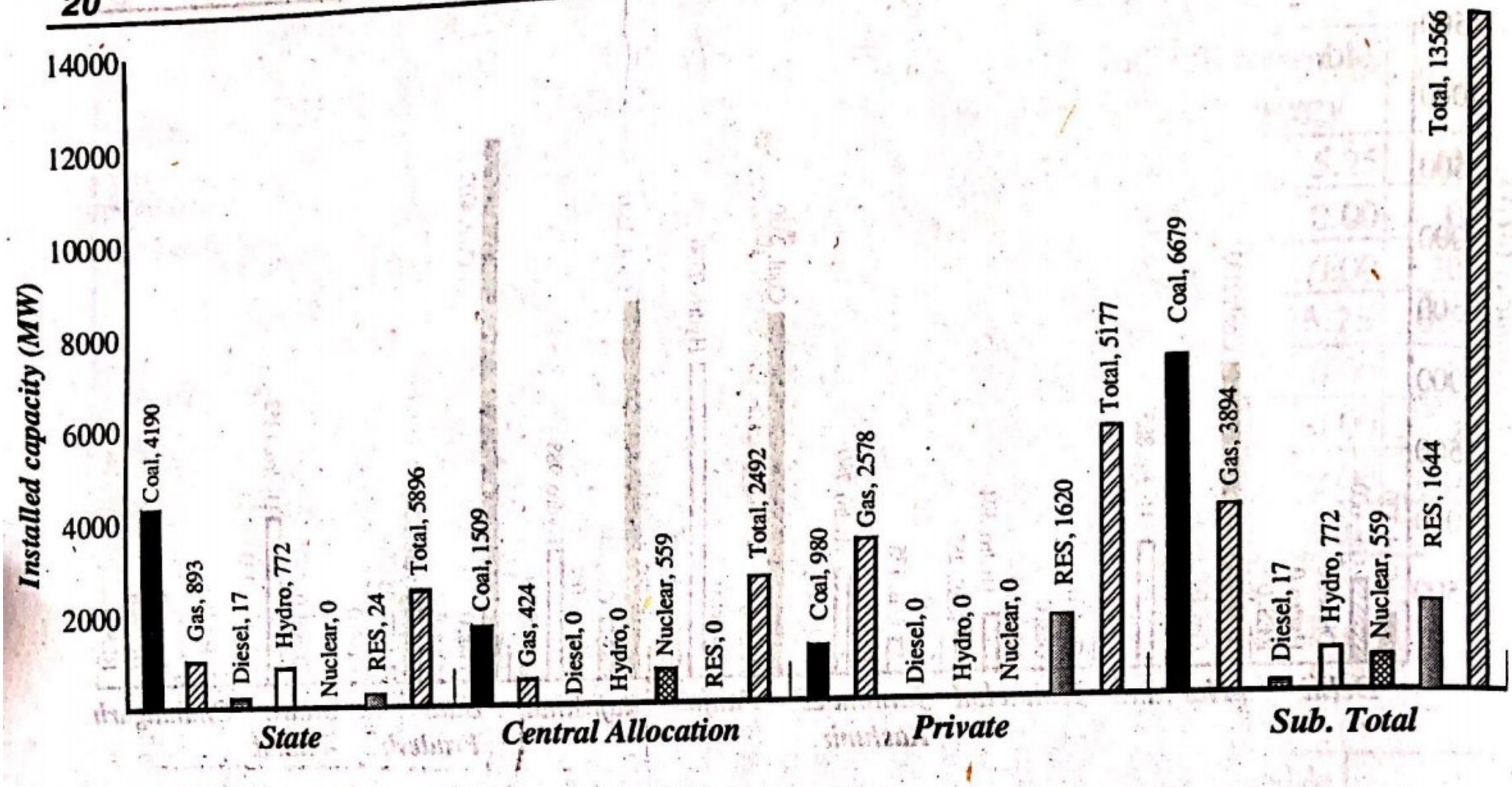


Fig. 1.6 Installed capacity (MW) of power in Gujarat (state, central allocation, private)

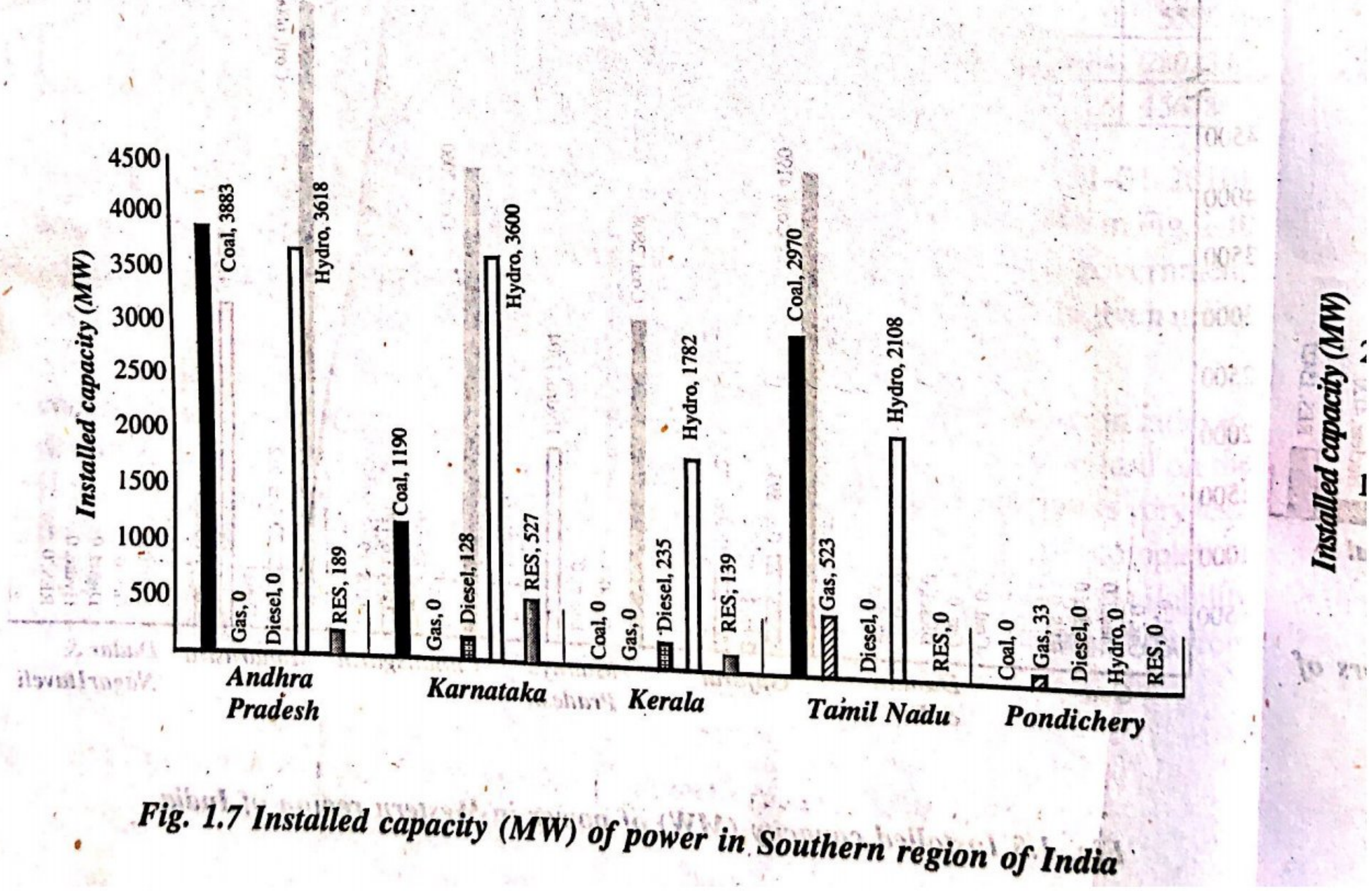


Fig. 1.7 Installed capacity (MW) of power in Southern region of India

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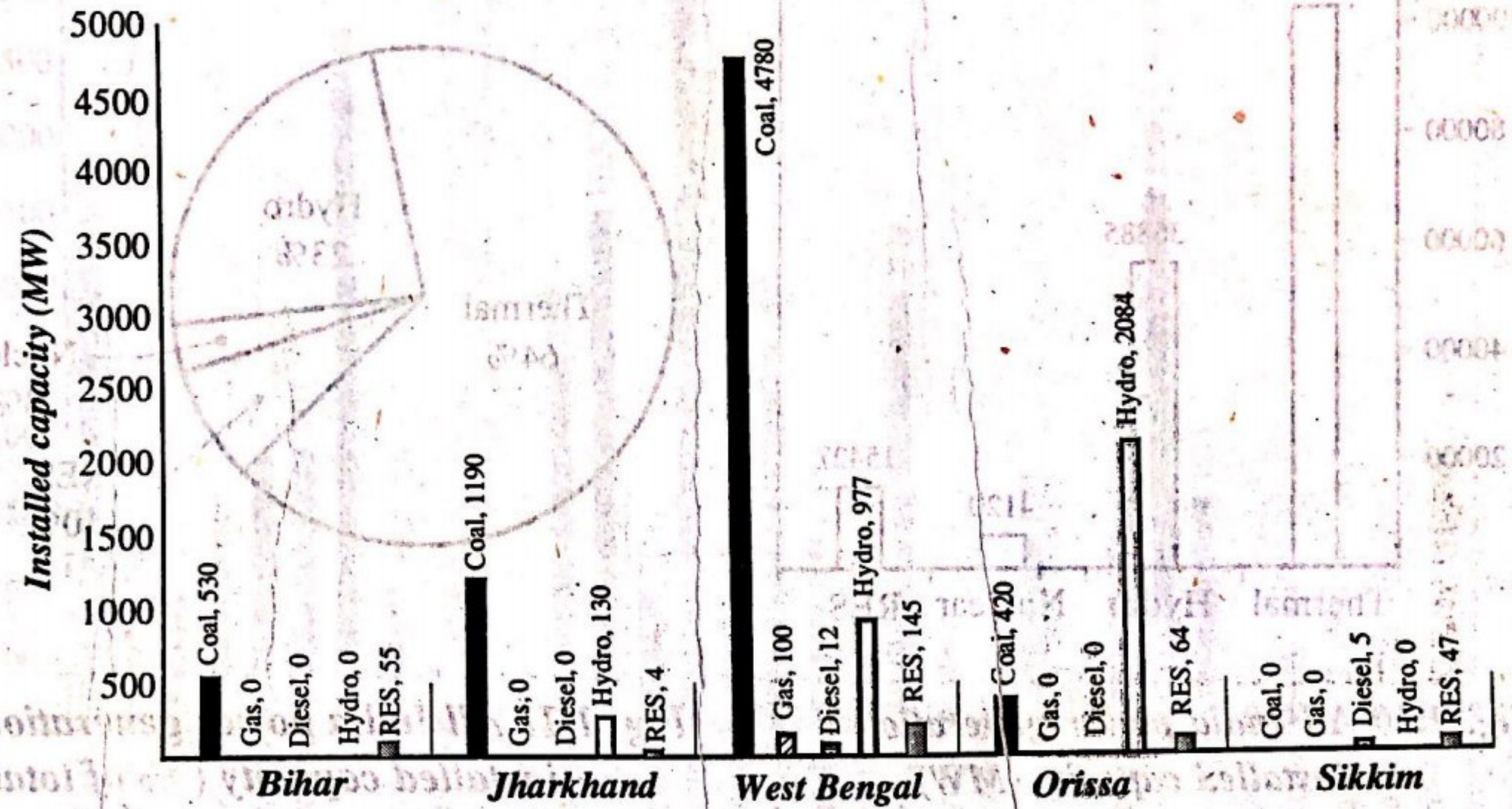


Fig. 1.8 Installed capacity ((MW) of power in Eastern region of India

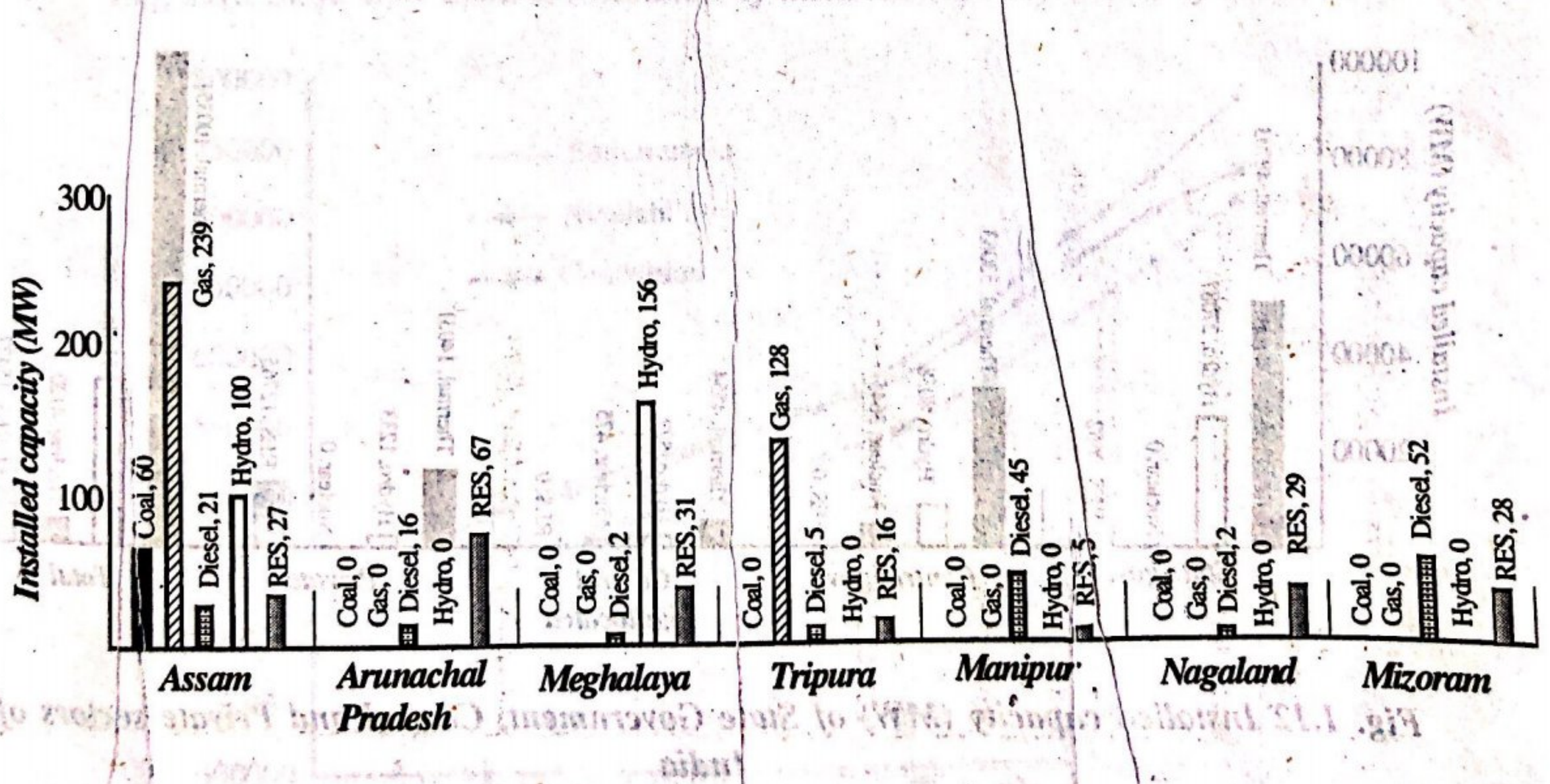


Fig. 1.9 Installed capacity (MW) of power in North-Eastern region of India

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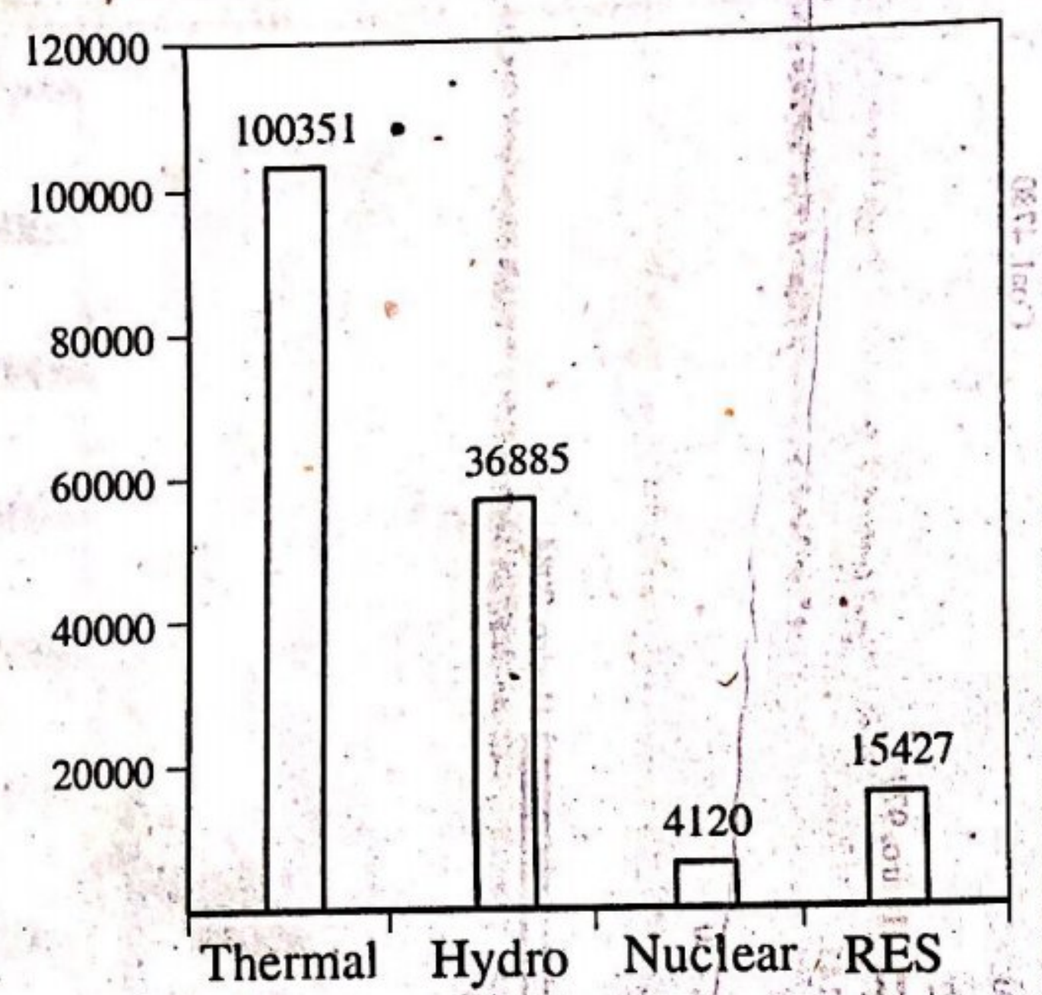


Fig. 1.10 All India power generation installed capacity (MW)

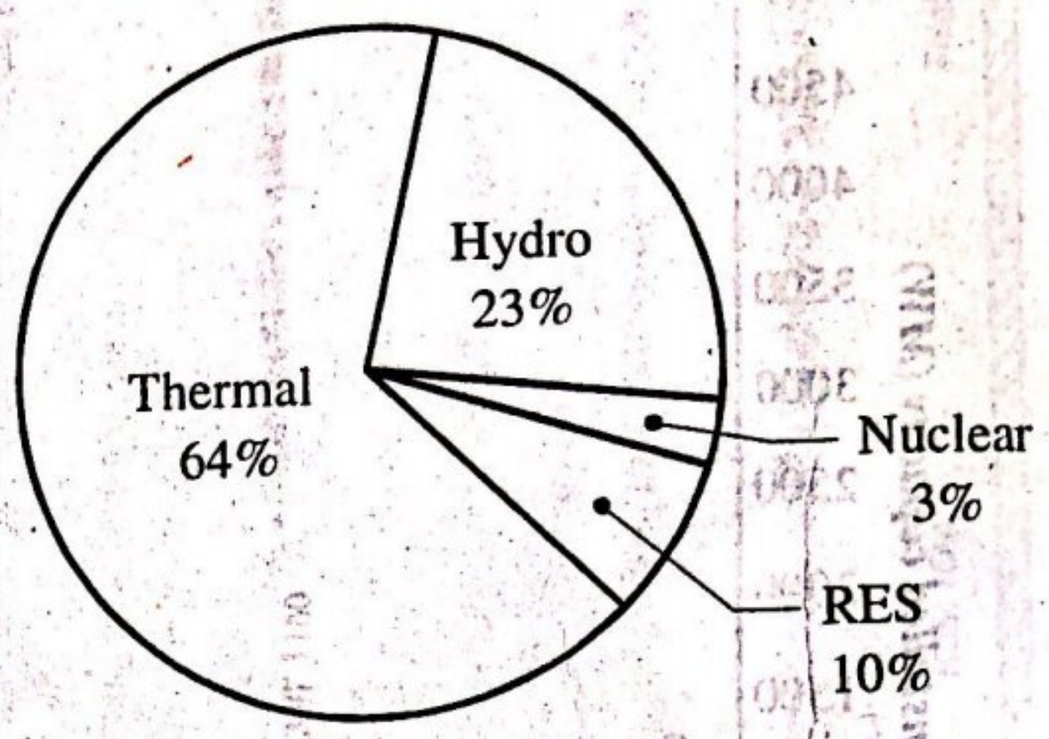


Fig. 1.11 All India power generation installed capacity (% of total)

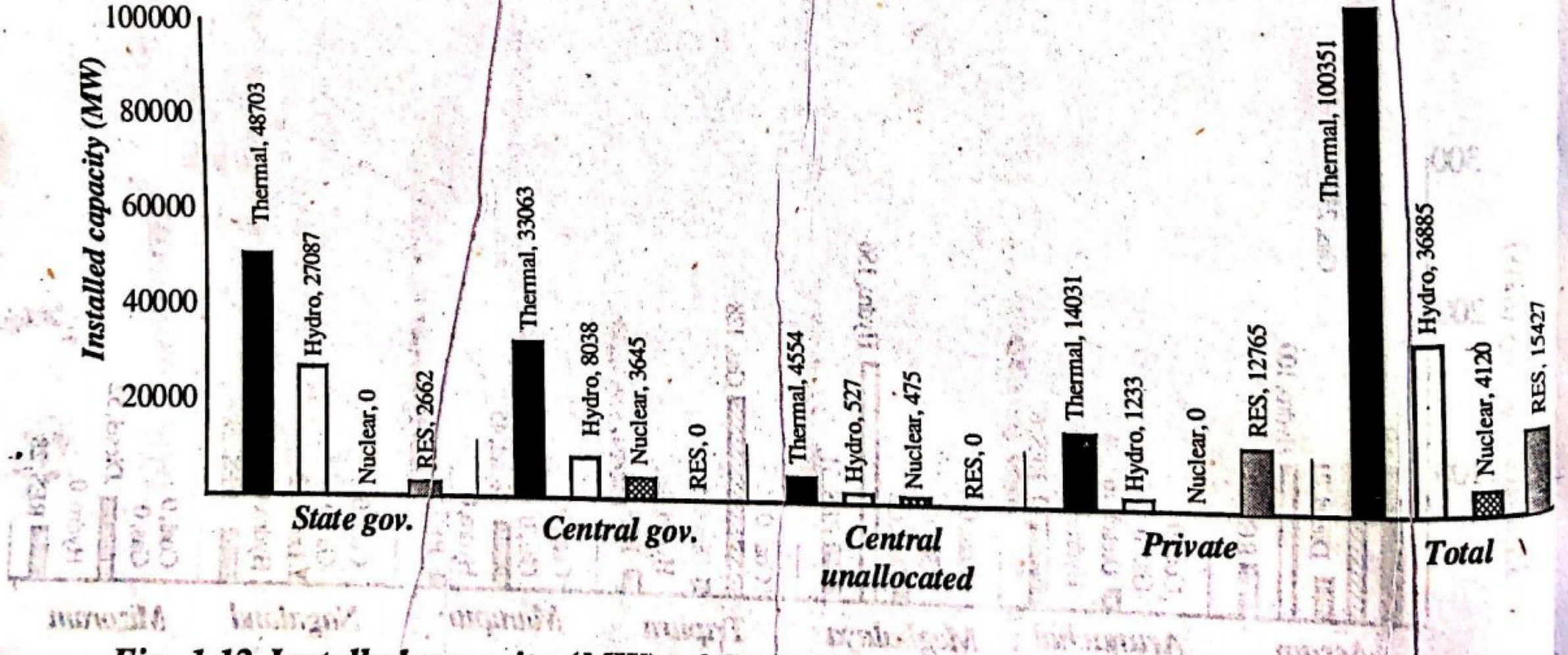


Fig. 1.12 Installed capacity (MW) of State Government, Central and Private sectors of India

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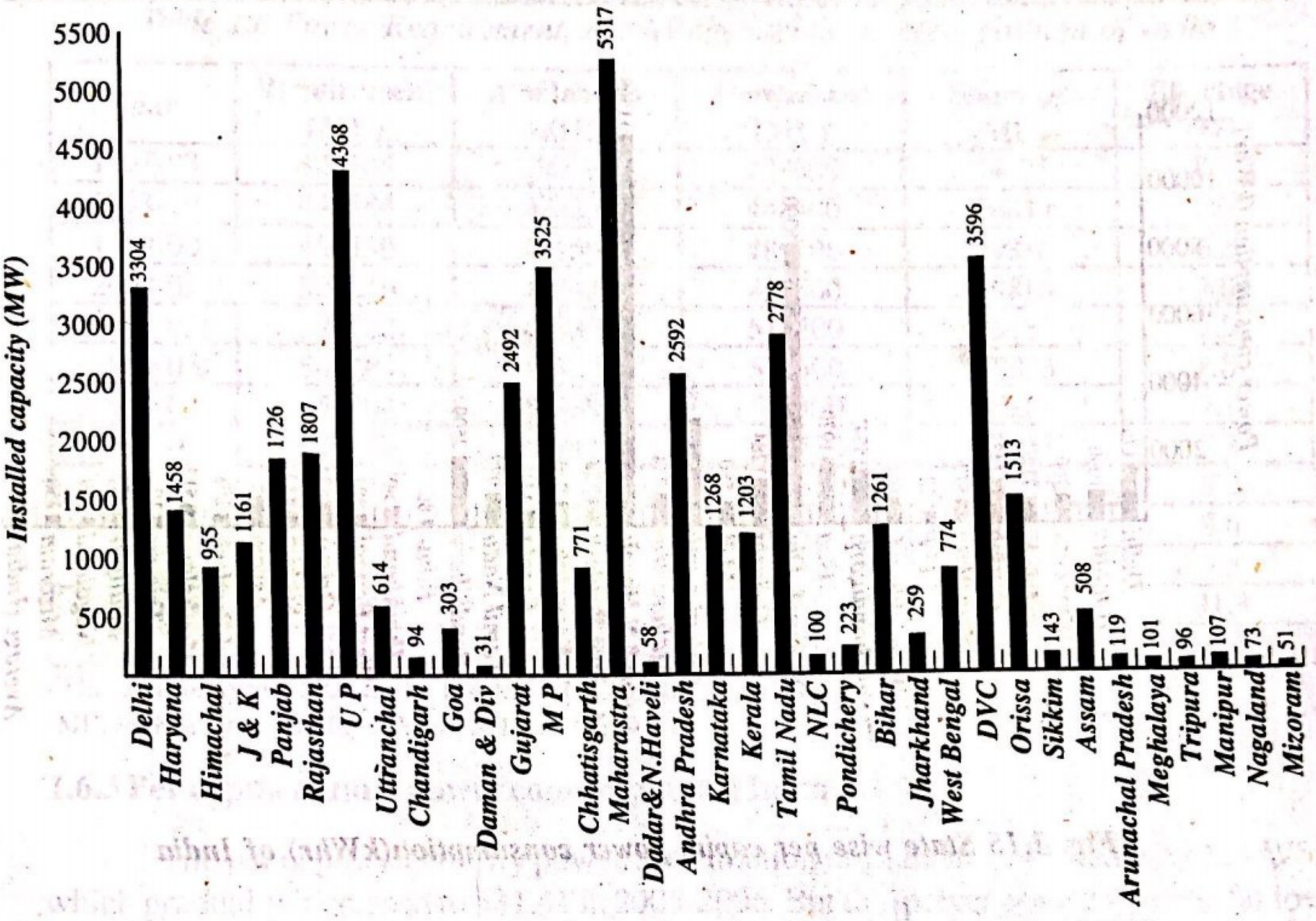


Fig. 1.13 State wise central allocation of installed capacity (MW) of India

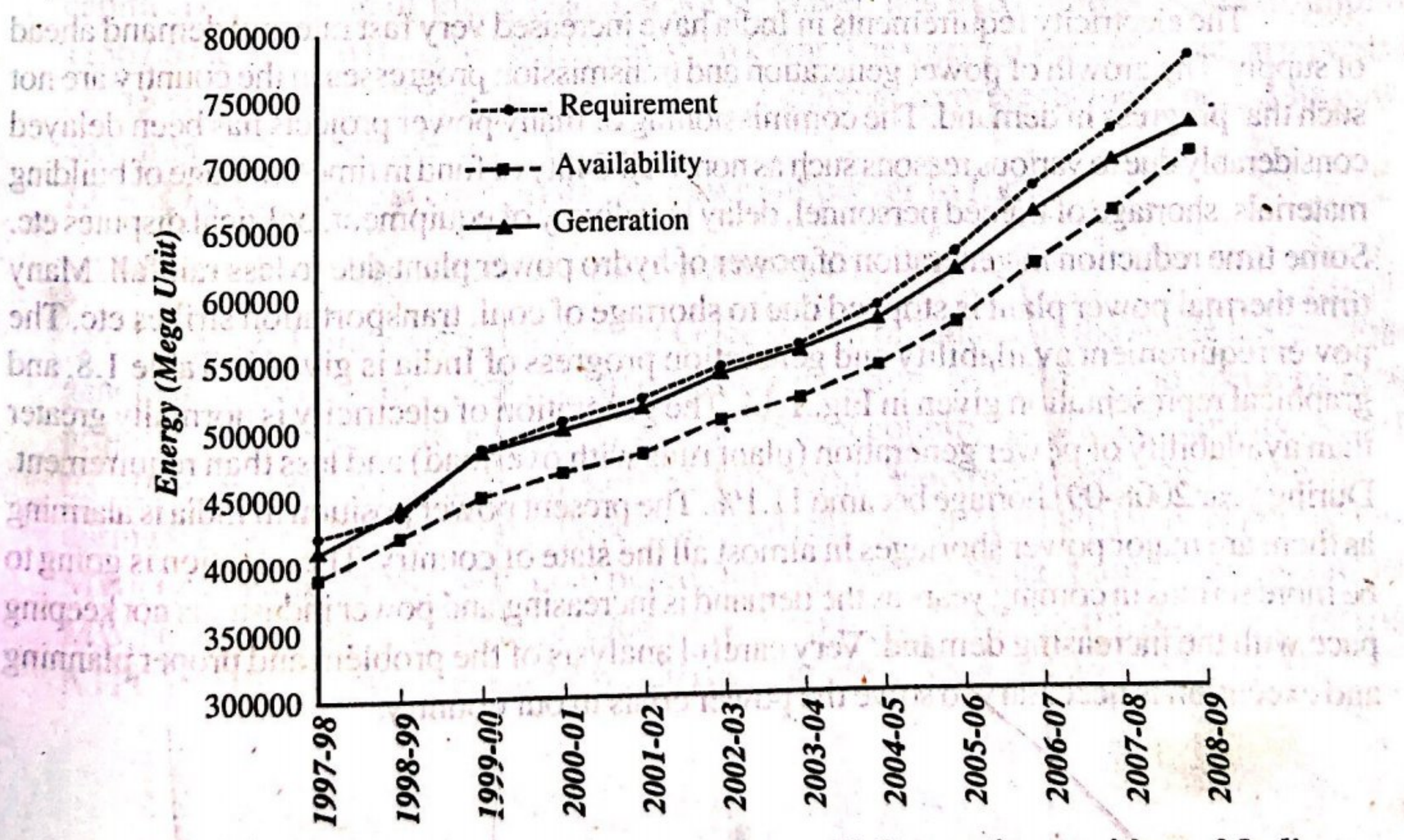


Fig. 1.14 Power Requirement, Availability and Generation position of India

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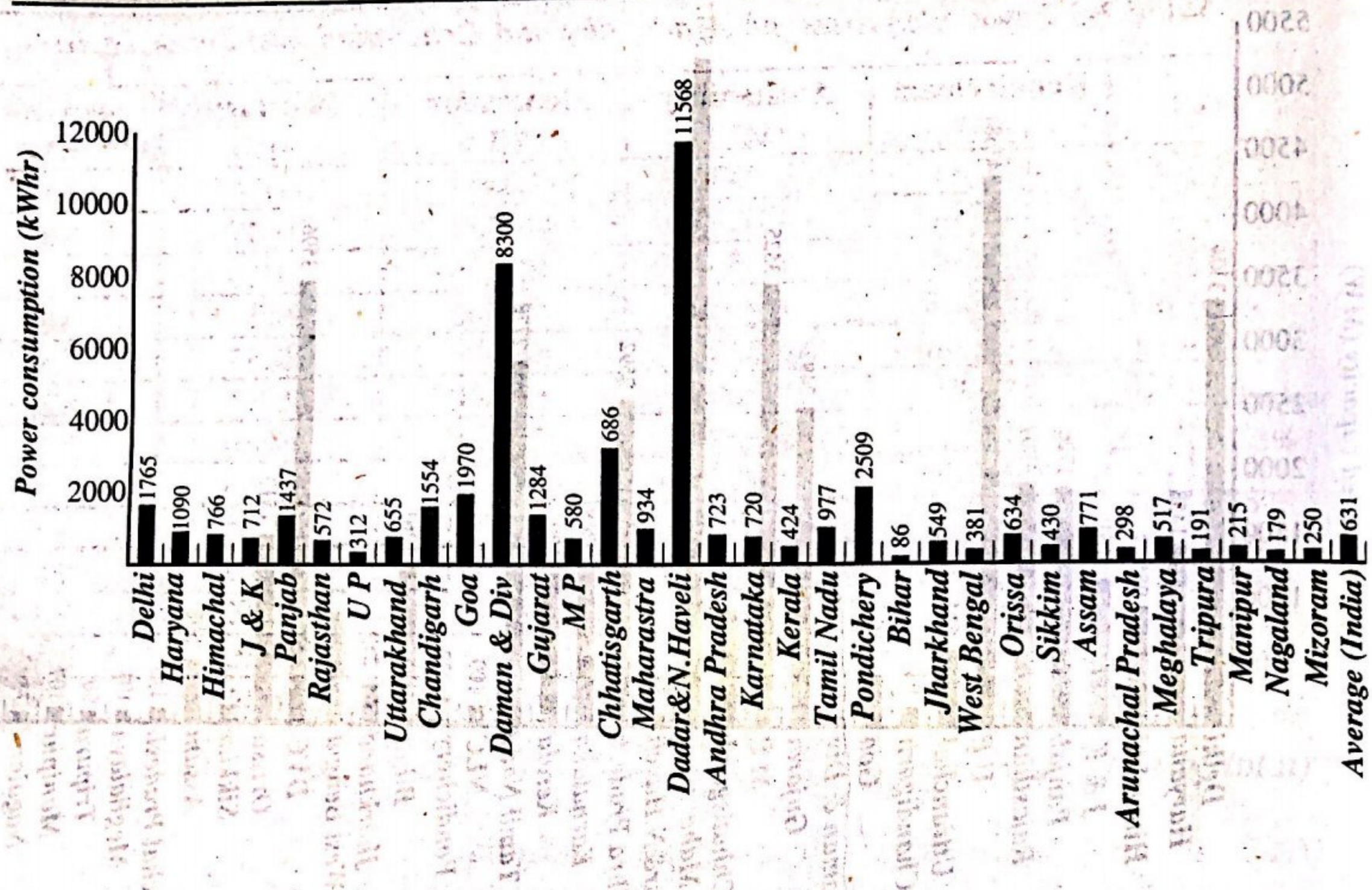


Fig. 1.15 State wise per capita power consumption(kWhr) of India

1.6.2 Power crisis in India

The electricity requirements in India have increased very fast rate and demand ahead of supply. The growth of power generation and transmission progresses in the country are not such that progress in demand. The commissioning of many power projects has been delayed considerably due to various reasons such as non availability of fund in time, shortage of building materials, shortage of trained personnel, delay in delivery of equipment, political disputes etc. Some time reduction in generation of power of hydro power plant due to less rainfall. Many time thermal power plant is stopped due to shortage of coal, transportation strikes etc. The power requirement availability and generation progress of India is given in Table 1.8. and graphical representation given in Fig. 1.14. The generation of electricity is normally greater than availability of power generation (plant runs with overload) and less than requirement. During year 2008-09 shortage became 11.1%. The present power position in India is alarming as there are major power shortages in almost all the state of country. The situation is going to be more serious in coming years as the demand is increasing and power industry is not keeping pace with the increasing demand. Very careful analysis of the problem and proper planning and execution is necessary to solve the power crisis in our country.

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Table 1.8 Power Requirement, Availability and Generation position of India

Year	Requirement (MU)	Availability (MU)	Generation (MU)	Shortage (MU)	Shortage (%)
1997-98	424505	390330	420600	34175	8.1
1998-99	446584	420235	448400	26349	5.9
1999-00	480430	450594	480700	29836	6.2
2000-01	507216	467400	499500	39816	7.8
2001-02	522537	483350	515200	39187	7.5
2002-03	545983	497890	531600	48093	8.8
2003-04	559264	519398	558300	39866	7.1
2004-05	591373	548115	587400	43258	7.3
2005-06	631554	578819	617500	52735	8.4
2006-07	690587	624495	662400	66092	9.6
2007-08	737052	664660	704400	72392	9.8
2008-09	777039	691038	723800	86001	11.1
2009-10	688171	620003	638100	68168	9.9

The data for year 2009-2010 is taken up to January, 2010

MU: Million Unit = 10 Lacs Units = 10 Lacs kWh

1.6.3 Per capita annual power consumption of India

The per capita electricity power consumption of India in 1947 was only 12 kWhr which gradually increased to 631.41 in 2005-2006. But this power consumption is far low compared with the developed countries. As shown in Table 1.9 state wise gross annual per capita consumption of India, Dadar & Nagar Haveli has maximum power consumption 11567.67 kWhr and Bihar has minimum 85.86 kWhr. The more power consumption represents the decent standard of living for the people. The graphical representation of per capita power consumption is given in Fig. 1.15.

1.6.4 Role of public sectors in development of power in India

NTPC - National Thermal Power Corporation :

NTPC Limited is a company of Government of India. It is the single largest power generator in India with comprehensive in house capability in building and operating power projects. NTPC generation capacity was 18.82% of country's installed capacity as on 31-03-2009. NTPC accounted for 28.60 % of the entire electricity generated in the country during the year 2008-09. Current operating capacity (as on 31-01-2010) of NTPC is 33970 MW comprising 28,095 MW coal based stations, 3955 MW gas based stations and 1920 MW hydro station. It also has a 2904 MW (coal based) under joint ventures. Details of NTPC installed capacity as on 30-01-2010 are given in Table 1.10 and Table 1.11.

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Table 1.9 State wise annual per capita power consumption of India (year 2005-2006)

State	Per capita power consumption (kWhr)
Haryana	1090.39
Himachal Pradesh	765.86
Jammu & Kashmir	711.01
Punjab	1436.79
Rajasthan	572.2
Uttar Pradesh	311.82
Uttarakhand	654.84
Chandigarh	1553.96
Delhi	1766.94
Gujarat	1283.77
Madhya Pradesh	580.34
Chhattisgarh	685.81
Maharashtra	934.43
Goa	1970.08
Daman & Diu	8300.12
D. & N Haveli	11567.67
Andhra Pradesh	723.1
Karnataka	720.43
Kerala	424.13
Tamil Nadu	976.81
Lakshadweep	368.29
Pondicherry	2509.25
Bihar	85.86
Jharkhand	548.74
Orissa	633.93
West Bengal	380.61
A.& N. Islands	407.77
Sikkim	429.81
Assam	170.65
Manipur	215.21
Meghalaya	517.54
Nagaland	179.34
Tripura	190.62
Arunachal Pradesh	297.66
Mizoram	250.15
Total (All India)	**631.41**

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Table 1.10 Installed capacity of NTPC's plants

(1) Thermal (coal) Based (owned by NTPC)		
City	**State**	**Installed Capacity**
Korba	Chhattisgarh	2,600
Ramagundam	Andhra Pradesh	2,600
Farakka	West Bengal	1,600
Vindhyachal	Madhya Pradesh	3,260
Rihand	Uttar Pradesh	2,000
Kahalgaon	Bihar	2,340
NCTPP, Dadri	Uttar Pradesh	2,310
Talcher Kaniha	Orissa	3,000
Unchahar	Uttar Pradesh	1,050
Talcher Thermal	Orissa	460
Simhadri	Andhra Pradesh	1,000
Tanda	Uttar Pradesh	440
Badarpur	Delhi	705
Sipat-II	Chhattisgarh	1,000
Sipat-I(erection phase)	Chhattisgarh	1,980
Bongaigaon(erection phase.)	Assam	750
Mouda(erection phase.)	Maharashtra	1,000
Total (coal based)		**28,095**
(2) GAS or liquid based (owned by NTPC)		
Anta	Rajasthan	413
Auraiya	Uttar Pradesh	652
Kawas	Gujarat	645
Dadri	Uttar Pradesh	817
Jhanor	Gujarat	648
Kayamkulam	Kerala	350
Faridabad	Haryana	430
Total (gas or liquid based)		**3,955**
(3) Hydro (owned by NTPC)		
Koldam (HEPP)	Himachal Pradesh	800
Loharinag Pala (HEPP)	Uttarakhand	600
Tapovan Vishnugad (HEPP)	Uttarakhand	520
Total (gas or liquid based)		**1,920**
Total (Coal based + Gas based+ Hydro)		**33,970**

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Table 1.11 Installed capacity of NTPC's Joint ventures plants

Coal Based (Owned by Joint Ventures)			
City	State	Joint Ventures	Installed Capacity
Durgapur	West Bengal	NSPCL	120
Rourkela	Orissa	NSPCL	120
Bhilai	Chhattisgarh	NSPCL	574
Nabinagar	Bihar	-	1980
Kanti	Bihar	-	110
Total			**2,904**

NHPC-National Hydro-electric Power Corporation :

NHPC limited (earlier known as National Hydroelectric Power Corporation Ltd.) is company of Government of India. NHPC was set up in 1975 and has now become the largest organisation for hydro power development of India, with capabilities of undertake all the activities from conceptualization to commissioning of Hydro projects. NHPC has so far commissioned 13 hydroelectric projects with an aggregate installed capacity of 5175 MW which includes 2 projects with total capacity of 1520 MW in joint venture with Government of Madhya Pradesh. Details of NHPC installed capacity as on 30-01-2010 are given in Table 1.12.

Table 1.12 Installed capacity of NHPC's Hydro Power Stations

Power Plant	State	Commissioned Capacity (MW)	Commissioned year
Baira siul	Himachal Pradesh	180	1981
Loktak	Manipur	105	1983
Salal-I	Jammu & Kashmir	345	1987
Tanakpur	Uttarakhand	120	1992
Chamera-I	Himachal Pradesh	540	1994
Salal-II	Jammu & Kashmir	345	1996
Uri-I	Jammu & Kashmir	480	1997
Rangit	Sikkim	60	1999
Chamera-II	Himachal Pradesh	300	2004
Indira Sagar	Madhya Pradesh	1000	2005
Dhauliganga-I	Uttarakhand	280	2005
Dul Hasti	Jammu & Kashmir	390	2007
Omkareshwar	Madhya Pradesh	520	2007
Teesta-V	Sikkim	510	2008
Total		**5175**	

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Under construction			
Kishenganga	Jammu & Kashmir	330	2016
Teesta Low Dam-III	West Bengal	1320	2011
Parbati-II	Himachal Pradesh	800	2013
Subansiri (Lower)	Assam	2000	2012
Chamera-III	Himachal Pradesh	231	2010
Teesta Low Dam-IV	West Bengal	160	2011
Parbati-III	Himachal Pradesh	520	2010
Nimmo-Bazgo	Jammu & Kashmir	45	2010
Chutak	Jammu & Kashmir	44	2011
Uri-II	Jammu & Kashmir	240	2011
Total (Under construction)		**5690**	
Total (Working + Under construction)		**10865**	

NPCIL - Nuclear Power Corporation of India Limited :

NPCIL (earlier Nuclear Power Board) is largest public sector of Government of India, was set up in 1987 to implement the nuclear power generation programme on commercial lines. It is responsible for designing, constructing, commissioning and operating all nuclear power reactors in the country. NPCIL has a commissioned 20 reactors with an installed capacity 4385 MWe and 5 reactors of total 3900 Mwe are under construction as given in Table 1.13.

Table 1.13 India's Nuclear Power Reactors under control of NPCIL

Reactor	State	Type of Reactor	MWe net, each	Commercial operation
Tarapur 1 & 2	Maharashtra	BWR	150	1969
Kaiga 1 & 2	Karnataka	PHWR	202	1999-2000
Kaiga 3 & 4	Karnataka	PHWR	202	2007
Kakrapar 1 & 2	Gujarat	PHWR	202	1993-95
Kalpakkam 1 & 2 (MAPS)	Tamil Nadu	PHWR	202	1984-86
Narora 1 & 2	Uttar Pradesh	PHWR	202	1991-92
Rajasthan 1	Rajasthan	PHWR	90	1973
Rajasthan 2	Rajasthan	PHWR	187	1981
Rajasthan 3 & 4	Rajasthan	PHWR	202	1999-2000
Rajasthan 5 & 6	Rajasthan	PHWR	202	2010
Tarapur 3 & 4	Maharashtra	PHWR	490	2006
Total (20)			**4385 MWe**	

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India's nuclear power reactors under construction				
Kudankulam 1 & 2	PWR (VVER)	1000	NPCIL	2011
Kalpakkam PFBR	FBR	500	Bhavini	2012
Kakrapar 3 & 4	PHWR	700	NPCIL	2015 -2016
Total (5)		**3900**		

REC- Rural Electrification Corporation :

REC was set up in 1969, with primary objective of promoting rural electrification by financing rural electrification schemes and rural electric corporatives in the states.

DVC - Damodar Valley Corporation :

DVC was established in 1948 under an Act of Parliament for unified development of Damodar Valley covering an area of 24,235 km² in Bihar and West Bengal. Main functions assigned to DVC are control of floods, irrigation, generation and transmission of power. The DVC has three thermal power stations at Bokaro, Chandrapura and Durgapur with total installed capacity of 1755 MW. It has four multi-purpose dams at Tilaiya, Maithon, Panchet, and Konar. There are three hydel power stations (total capacity 144 MW). DVC has also set up three gas turbine unit (total capacity 90 MW) at Maithon.

Table 1.14 Statewise India's Thermal Power Plants (Data taken on: 22 September 2010)

Gujarat Thermal Power Plants			
Power Station	**Location**	**District**	**Installed Capacity(MW)**
Ukai TPS	Ukai dam	Tapi	850
Gandhinagar TPS	Gandhinagar	Gandhinagar	870
Wanakbori TPS	Wanakbori	Kheda	1470
Sikka TPS	Jamnagar	Jamnagar	240
Dhuvaran Thermal Power Plant	Khambhat	Anand	220
Kutch Lignite TPS	Panandhro	Kutch	290
Surat Lignite TPS	NaniNaroli	Surat	500
Akrimota Lignite Power Plant	ChherNani	Kutch	250
Mundra TPP	Mundra	Kutch	660
Sabarmati TPS	Ahamadabad	Ahamadabad	400

Punjab Thermal Power Plants			
Power Station	**Location**	**District**	**Installed Capacity(MW)**
Guru Nanak dev TP	Bathinda	Bathinda	440
Guru Hargobind TP	LehraMohabb	Bathinda	920
Guru Gobind SSTP	Ghanauli	Rupnagar	1260

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Andhra Pradesh Thermal Power Plants			
Power Station	**Location**	**District**	**Installed Capacity(MW)**
Ramagundam B TPS	Ramagundam	Karimnagar	62.5
Kothagudem TPS	Paloncha	Khammam	720
Kothagudem V stage TPS	Paloncha	Khammam	500
DrNarlaTatarao TPS	Ibrahimpatnam	Krishna	1760
Rayalaseema TPS	Cuddapah	YSR	840
Kakatiya TPS	Chelpur	Warangal	500
Ramagundam STPS	Jyothi Nagar	Karimnagar	2600
Simhadri STPS	Simhadri	Visakhapatna	1000

Chhattisgarh Thermal Power Plants			
Power Station	**Location**	**District**	**Installed Capacity(MW)**
Korba STPP	JamaniPalli	Korba	2100
Sipat TPP	Sipat	Bilaspur	1000
Jindal Mega PP	Tamnar	Raigarh	1000
LancoAmarkantak TPP	Pathadi	Korba	600

Orissa Thermal Power Plants			
Power Station	**Location**	**District**	**Installed Capacity(MW)**
IB Thermal pp	Banharpali	Jharsuguda	420
Talcher STPS	Kaniha	Angul	3000
Talcher TPP	Talcher	Angul	460

Tamilnadu Thermal Power Plants			
Power Station	**Location**	**District**	**Installed Capacity(MW)**
North Chennai TPS	Athipattu	Thiruvallore	630
Ennore TPS	Ennore	Chennai	450
Mettur TPS	Metturdam	Salem	840
Tuticorin TPS	Tuticorin	Tuticorin	1050
Neyveli TPS 1	Neyveli	Cuddalore	1020
Neyveli TPS 2	Neyveli	Cuddalore	1470
Neyveli Zero Unit	Neyveli	Cuddalore	250

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Madhya Pradesh Thermal Power Plants			
Power Station	**Location**	**District**	**Installed Capacity(MW)**
Satpura TPS	Sarni	Betul	1017.5
Sanjay Gandhi TPS	Birsinghpur	Umaria	1340
Amarkantak TPS	Chachai	Anuppur	450
Vindhyachal STPS	Vidhya Nagar	Sidhi	3260

Uttar Pradesh Thermal Power Plants			
Power Station	**Location**	**District**	**Installed Capacity(MW)**
Obra Thermal Power Station	Obra	Sonebhadra	1322
Anpara Thermal Power Station	Anpara	Sonebhadra	1630
Panki Thermal Power Station	Panki	Kanpur	210
Parichha Thermal Power Station	Parichha	Jhansi	640
Harduaganj Thermal Power	Harduaganj	Aligarh	220
Singrauli Super Thermal Power	Shaktinagar	Sonebhadra	2000
Rihand TPP	Rihand Nagar	Sonebhadra	2000
National Capital TPP	Vidyutnagar	GautamBudh	1820
Feroz Gandhi Unchahar TPP	Unchahar	Raebareli	1050
Tanda TPP	Vidyutnagar	Ambedkar	440
Rosa Thermal Power Plant	Rosa	Shahjahanpur	600

Bihar Thermal Power Plants			
Power Station	**Location**	**District**	**Installed Capacity(MW)**
Barauni TPP	Barauni	Begusarai	310
Muzaffarpur TPP	Kanti	Muzaffarpur	220
Kahalgaon STPS	Kahalgaon	Bhagalpur	2340

Haryana Thermal Power Plants			
Power Station	**Location**	**District**	**Installed Capacity(MW)**
DeenbandhuChhotu Ram TPP	Yamunanagar	Yamunanagar	600
Panipat TPS 1	Assan	Panipat	440
Panipat TPS 2	Assan	Panipat	920
Faridabad Thermal Power Station	Faridabad	Faridabad	55
Rajiv Gandhi Thermal Power	Khedar	Hisar	600

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Maharashtra Thermal Power Plants			
Power Station	**Location**	**District**	**Installed Capacity(MW)**
Koradi TPS	Koradi	Nagpur	1040
Nashik TPS	Nashik	Nashik	880
Bhusawal TPS	Deepnagar	Jalgaon	470
Paras TPS	Vidyutnagar	Akola	555
Parli TPS	Parli-Vaijnath	Beed	1170
Kaparkheda TPS	Kaparkheda	Nagpur	840
Chandrapur STPS	Chandrapur	Chandrapur	2340
Trombay Thermal Power Station	Trombay	Mumbai	1400
Dahanu Thermal Power Station	Dahanu	Thane	500
WardhaWarora Power Plant	Warora	Chandrapur	135

Rajasthan Thermal Power Plants			
Power Station	**Location**	**District**	**Installed Capacity(MW)**
Suratgarh STPS	Suratgarh	Suratgarh	1500
Kota STPS	Kota	Kota	1240
Giral Lignite TPS	Thumbli	Barmer	250
Chhabra STPP	Mothipura	Baran	500
Barsingsar Lignite Power Plant	Barsingsar	Bikaner	125
VS Lignite Power Plant	Gurha	Bikaner	125

West Bengal Thermal Power Plants			
Power Station	**Location**	**District**	**Installed Capacity(MW)**
Kolaghat TPS	Mecheda	East	1260
Bakreswar TPS	Suri	Birbhum	1050
Sagardighi TPS	Manigram	Murshidabad	600
Durgapur TPP	Durgapur	Bardhaman	690
Farakka STPS	Nagarun	Murshidabad	1600
Durgapur Thermal Power Station	Durgapur	Bardhaman	350
Mejia Thermal Power Station	Durlavpur	Bankura	1340
Budge Budge Thermal Power	Achipur	South 24	750

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1.7 Comparison of various power plants [Dec. '13]

Sr. No.	Particulars	Steam Power Plant	Hydro-electric Power Plant	Diesel Power Plant	Gas Turbine Power Plant	Nuclear Power Plant
1	Major site selection factors	It can be located near the load centre with consideration of availability of fuel and water, cost of land, transportation facilities etc.	It should be located where huge amount of water available with sufficient head, large area available with low cost and availability of transport facilities.	It can be installed anywhere.	This plant can also be installed anywhere.	It must be located far away from dense populated area but near the load centre as possible.
2	Flexibility of site	It is more flexible as compared to hydro-electric power plant.	Less flexibility for location because it requires huge amount of water with sufficient head. Normally plant is located far away from the load centre.	Highest flexibility for site selection compared to other power plant.	More flexible compared to steam, hydro and nuclear power plants.	More flexible compared to steam and hydro power plants but radio-active pollution is main consideration.
3	Space requirements	Less than hydro but larger than other power plants.	Very large, maximum	Less, minimum	Larger than diesel but less than other power plants.	Less than steam and hydro but larger than diesel and gas power plants.
4	Fuel	Coal, oil	No fuel , water is source of power	Diesel	Gas	Uranium, thorium
5	Auxiliaries requirements	Large	Less	Very less	Moderate	Large
6	Initial costs	Lower than hydro and nuclear power plants.	High as compared to steam power plant.	Minimum	Low as compared to steam power plant.	Very high as compared to other power plants.
7	Operating costs	Medium	Minimum (practically nil)	Very high	Quite high	Very low
8	Fuel transportation costs	High in case of coal mines far	Nil because no fuel requirement.	Higher than gas but lower than steam	Moderate	Very low because fuel requirement

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Sr. No.	Particulars	Steam Power Plant	Hydro-electric Power Plant	Diesel Power Plant	Gas Turbine Power Plant	Nuclear Power Plant
8	Fuel transportation costs	High in case of coal mines far away from site.	Nil because no fuel requirement.	Higher than gas but lower than steam power plant.	Moderate	Very low because fuel requirement is very small
9	Maintenance costs	Higher as compared to hydro and diesel power plant because skilled operating staffs are required.	Very low because less operating staffs are required.	Very low because less operating staffs are required.	Comparatively low	Very high because very skilled operating staffs are required.
10	Transmission and distribution costs	Moderate because plant is not much away from load centre hence short transmission lines are required.	Very high, since it is far away from load centre hence long transmission lines are required.	Practically nil because it is located at load centre	Moderate	Quite low
11	Overall efficiency	30 - 40%	80 - 85%	30 - 35%	30 - 50%	30 - 40%
12	Plant life	20-30 years	30-50 years	15-20 years	15-20 years	30-50 years
13	Commissioning and erection period	3-5 years	10-15 years	0.5-2 years	1-2 years	30-50 years
14	Starting time	2-3 hours	5-10 minutes	5-10 minutes	10-20 minutes	2-3 hours
15	Reliability	Less reliable	More reliable	Less reliable	Reliable	Reliable
16	Pollution	High air pollution	Pollution free	Less pollution compared to steam power plant	Less pollution compared to steam power plant	No air pollution, but special care is required for waste disposal.
17	Field of application	Most economical and suitable if plant located near to coal mines and water source. Normally it is used for base load.	Most economical and suitable where sufficient water source available with sufficient head. Normally it is used for base load.	It is suitable for small power requirements for emergency or used where other power plant is not suitable. Normally it is used as a mobile plant, stand by plant, or emergency plant.	It is suitable where natural gas is available with less cost. Normally is used as a stand by unit or peak load unit with steam power plant.	It is used where neither water nor coal as a source of power is available.

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Exercises

1.1 Write explanatory note on layout of a modern thermal power station including the main circuit and path of flow.

1.2 Draw general layout of modern steam power plant label major component and state function of each component. **[Dec. 2010, 07 Marks]**

1.3 Write short notes on layout of thermal power plant.

1.4 Give the layout of modern thermal power station including major circuits/ paths of flow of coal, air & flue gases, condensate & steam and cooling water. Label the major equipments.

1.5 Briefly explain the factors to be considered for site selection of a modern thermal power plant.

1.6 State the various types of conventional power plants.

1.7 What are the basic resources in India for power generation ?

1.8 Explain the present power position of India ?

Objective Type Questions

1. When capacity factor of power plant is greater them 57% , then power plant is known as _____

(a) central power station (b) captive power plant

(c) Base load plant (d) Peak load plant

2. Which of the following is most common source of power in India.

(a) coal (b) oil (c) Uranium (d) gas

3. The public sector unit has a inhouse capability in building and operating power project in India is

(a) NPCIL (b) NHPC (c) NTPC (d) REC

4. The public sector unit associated with the nuclear power generation in India is

(a) NPCIL (b) NHPC (c) NTPC (d) REC

5. What are the main functions assigned to Damodar Valley corporation.

(a) power generation and transmission (b) control of floods

(c) irrigation (d) all of these

6. Which of the following is largest organisation for hydroelectric power project

(a) NPCIL (b) NHPC (c) NTPC (d) REC

7. Steam power plants work closely on

(a) Carnot cycle (b) Brayton cycle

(c) Rankine cycle (d) Auto cycle

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8. Operating cost of steam power plant is less than
   (a) Nuclear power plant (b) Diesel power plant
   (c) Hydro electric power plant (d) none of these
9. The efficiency of the thermal power plant is of the order of
   (a) 60% (b) 45% (c) 30% (d) 20%
10. Which types of power plant has highest contribution to generate the power in India.
   (a) Thermal (b) Hydro (c) Nuclear (d) RES

Answers

1. (c) **2.** (a) **3.** (c) **4.** (a) **5.** (d) **6.** (b) **7.** (c) **8.** (b) **9.** (c) **10.** (a)

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GTU Study Materials ☑

Chapter 2
High Pressure Boilers

2.1 Introduction to high pressure boilers
2.2 Unique features of high pressure boilers
2.3 La-Mont boiler
2.4 Benson boiler
2.5 Loeffler boiler
2.6 Schmidt-Hartmann boiler
2.7 Supercritical boiler
2.8 Supercharged boiler
2.9 Fluidized Bed Combustion (FBC) system
2.9.1 Atmospheric FBC boilers
2.9.2 Pressurized FBC boiler
2.10 Heating surface and their location in the boiler
2.11 Superheaters
2.12 Reheaters
2.13 Economizers
2.14 Air pre-heaters
2.15 Steam temperature or superheat control
2.16 Corrosion in boilers and its prevention

2.1 Introduction to high pressure boilers

In applications where steam is needed at 30 bar and steam generating capacity less than 30,000 kg/hr, the fire tube boilers are considerably cheaper than the water tube boilers. But the steam requirement is above 30 bar pressure and higher rate, the fire tube boilers will be very bulky and difficult to transport to the site. The water tube boiler are not facing this problem e.g. Babcock and Wilcox boiler has maximum working pressure 42 bar while Lancashire's 16 bar. Therefore, water tube boiler is generally preferred for high pressure and high steam generating capacity whereas fire tube boiler for low pressure and low capacity.

In the power plant, it is necessary to generate steam at a higher rate, high pressure and with higher efficiency. This requirement is fulfill by high pressure boilers. The modern high pressure boilers are generate steam at rate 30 to 650 tones/hr and above with a pressure up to 160 bar and maximum steam temperature of about 540° C for power generation applications.

The various type of high pressure boilers commonly used are:

(1) La-mont boiler, (2) Loeffler boiler, (3) Benson boiler, (4) Schmidt-Hartmann boiler, (5) Velox boiler, (6) Supercritical boiler, (7) Supercharged boiler, (8) Fluidized bed combustion boilers etc.

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2.2 Unique features of high pressure boilers

(1) Method of water circulation

In case of natural circulation boiler (low pressure boiler), flow of water/steam takes place due to density difference of hot water and cold water as shown in Fig.2.1. With an increase in boiler pressure the density difference reduces, at critical pressure (221.05 bars) steam and water having same density. Hence natural circulation is not useful for high pressure boiler specially critical and supercritical boiler. Also, heat transfer point of view it is simpler to use high water velocities rather than high gas velocities. Further, slow circulation of water causes film boiling and hence steam bubbles remains contact with metal tube which increases the resistance to heat flow.

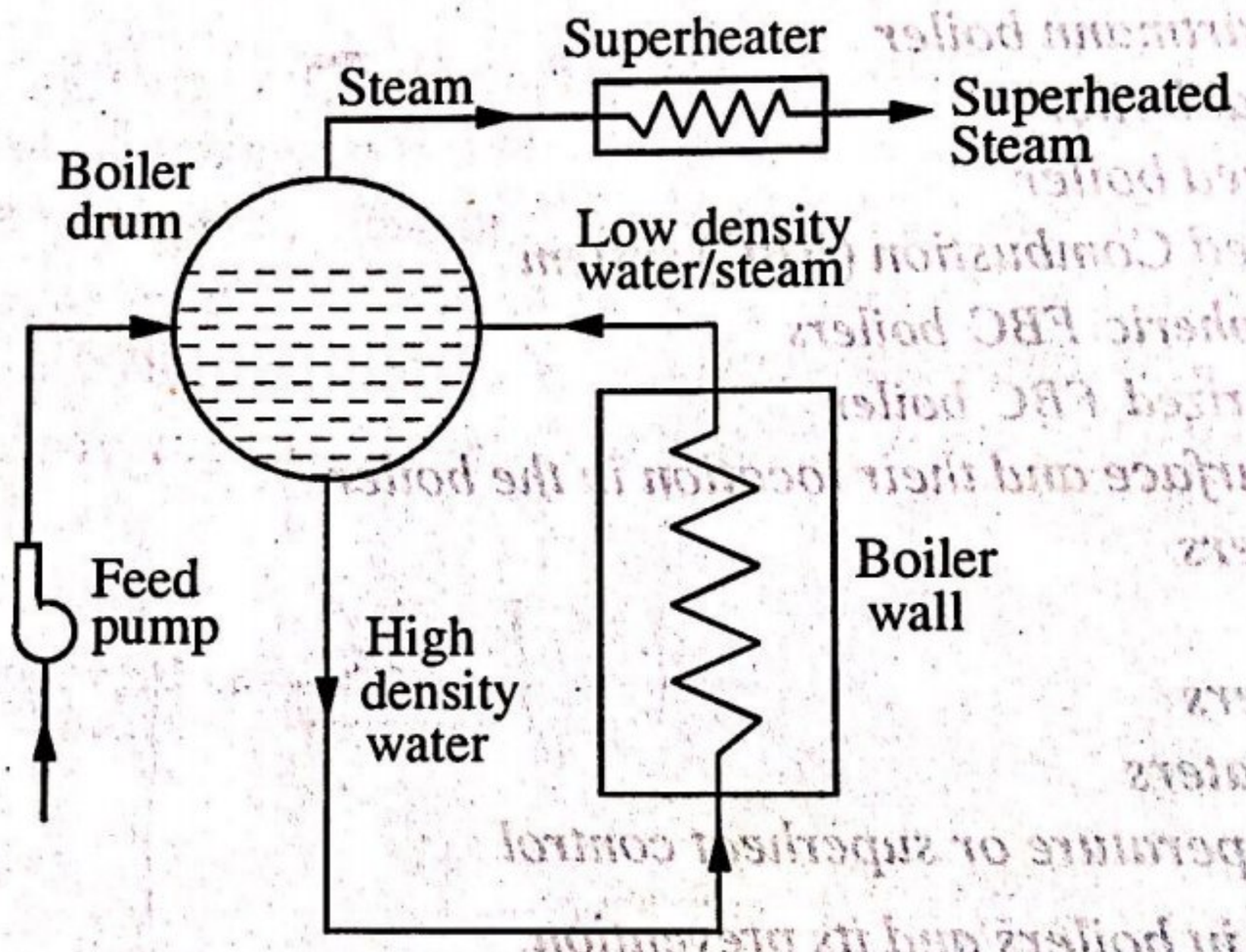


Fig 2.1 Natural circulation boiler

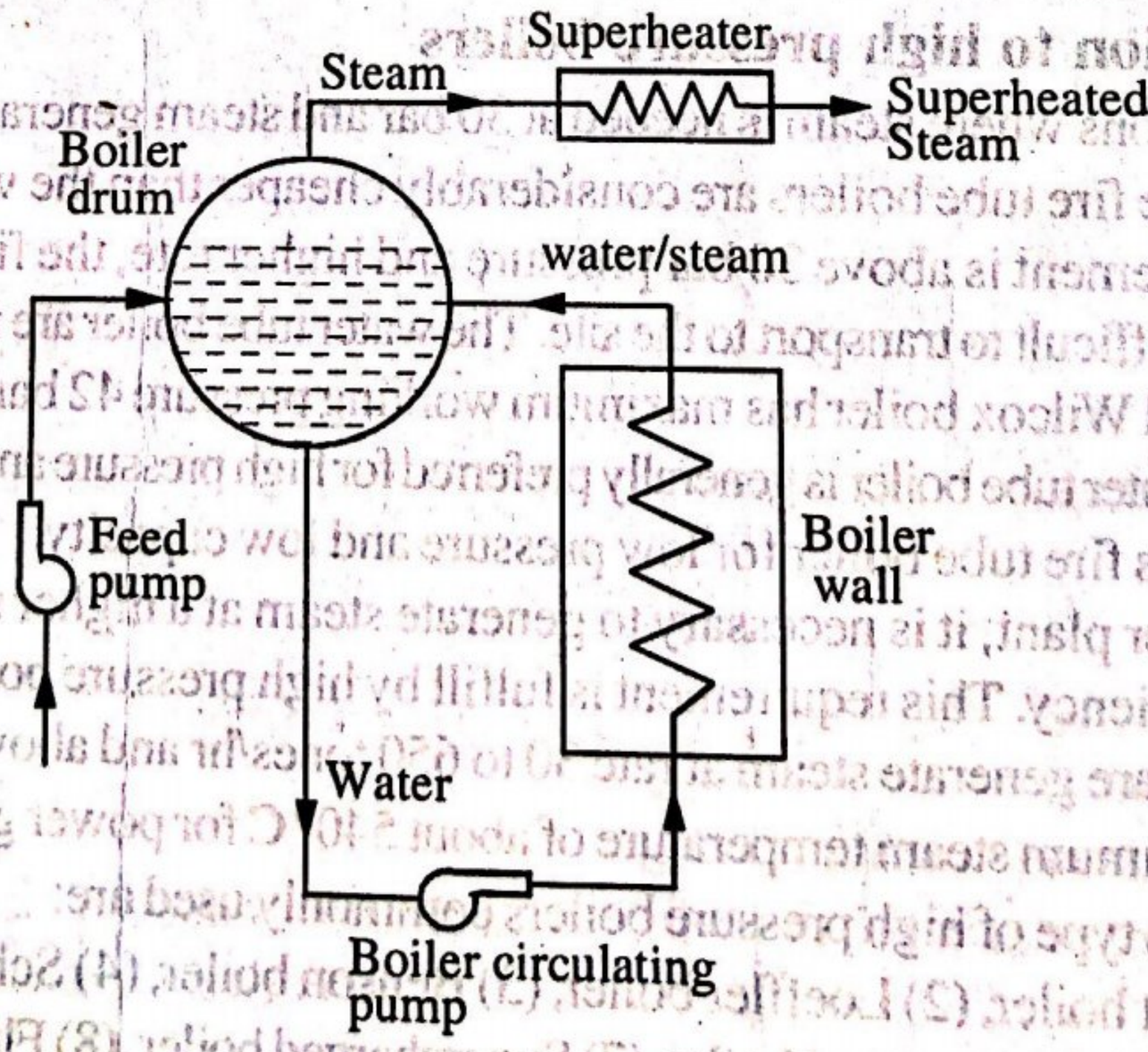


Fig 2.2 Forced circulation boiler

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In all modern high pressure boiler plants, the water circulation is maintained with the help of boiler circulating pump which forces the water through the boiler as shown in Fig.2.2. Due to high velocity of water in forced circulation the heat transfer rate is increased. Hence, evaporative capacity of the boiler can be increased and size of boiler is reduced by using smaller diameter tubes.

(2) Type of tubing

The most of high pressure boilers are water tube boilers. If flow takes place through one long continuous tube, the large pressure drop takes place due to friction. This is reduced by arranging the flow to pass through parallel set of tubes. This arrangement in high pressure boilers reduces the pressure loss and provides better control over the quality of the steam.

In the force circulation boiler it is possible to prevent film boiling. But, at boiler pressure above 200bar the effectiveness of drum in separating the saturated steam from water is reduced. This negative can be eliminated by using once through boiler. In once through boiler as shown in the Fig. 2.3 water is fed to the boiler at one end and superheated steam discharged through the other end. Hence, entire operation of feed water heating, steam formation (evaporation) and superheating are taken place in a single tube without use of drum.

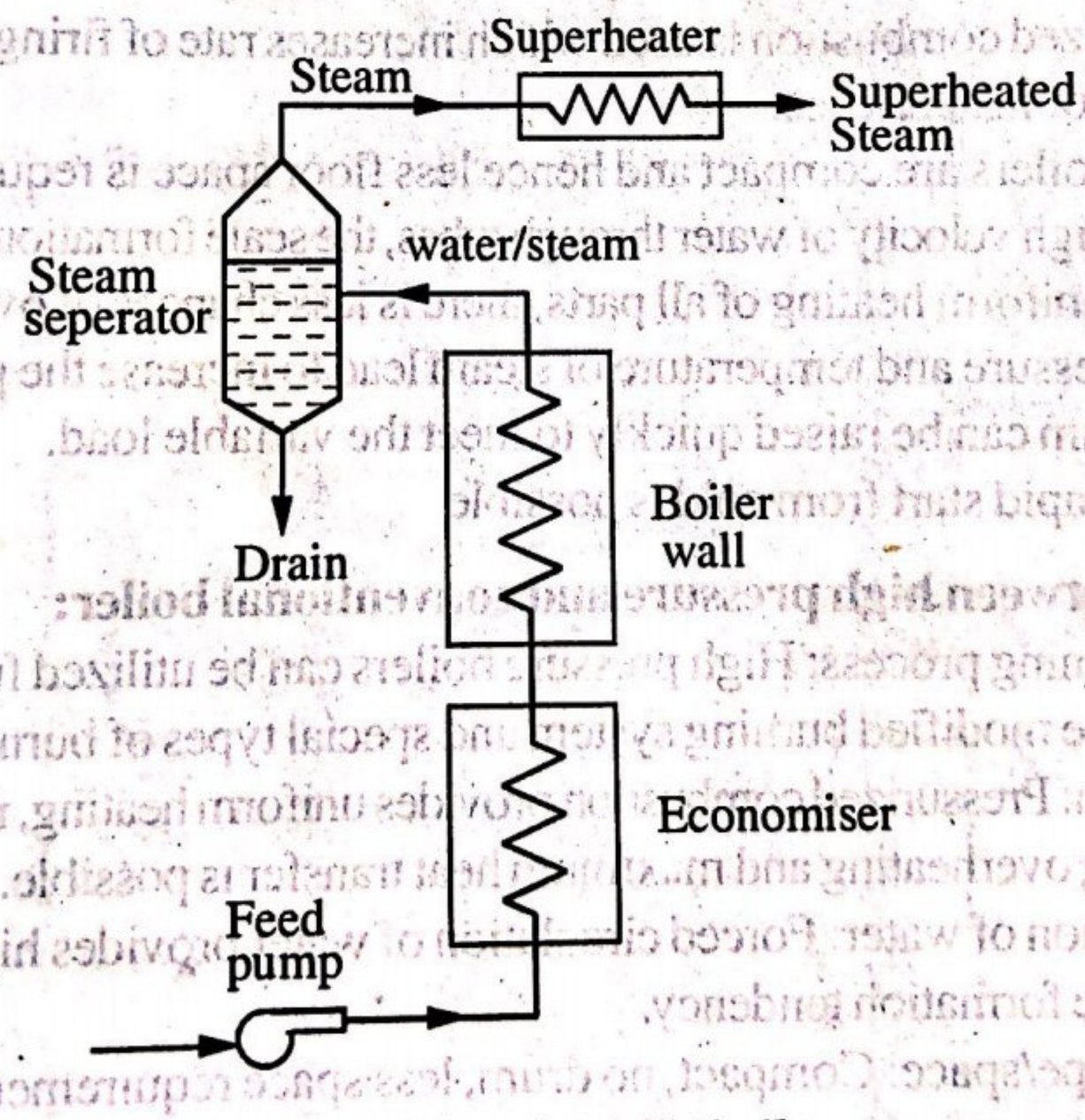


Fig 2.3 Once through boiler

(3) Pressurized combustion

In order to increase the rate of heat release in the boiler furnaces it is essential to increase the rate of firing of the fuel. Hence it is necessary to use pressurized combustion systems.

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(4) **Improved methods of heating**

In high pressure boilers, one or more of the following methods can be adopted to increase heat transfer.

(i) Saving of latent heat of evaporation of water at pressure above critical pressure.

(ii) Heating of water by mixing the superheated steam.

(iii) Velocity of hot gas and water are high and hence heat transfer rate from gas to water is increased.

(iv) Utilization of all three mode of heat transfer as conduction, convection and radiation. In modern high pressure boiler about 35% of total heat transfer takes place by radiation.

(5) **Compactness:** In high pressure boiler, large number of small size tubes and number of pass are used which increases the heat transfer area, hence size and weight of boiler is reduced.

Advantages of high pressure boiler over conventional boiler:

(1) Due to forced circulation of water, evaporative capacity of boiler is increased and size of drum is reduced.

(2) Smaller diameter tubes are used, which increases heat transfer rate and reduces fuel consumption.

(3) Pressurized combustion is used which increases rate of firing of fuel thus increasing the rate of heat release.

(4) These boilers are compact and hence less floor space is required.

(5) Due to high velocity of water through tubes, the scale formation tendency is eliminated.

(6) Due to uniform heating of all parts, there is less chances of overheating.

(7) High pressure and temperature of steam lead to increase the plant efficiency.

(8) The steam can be raised quickly to meet the variable load.

(9) A very rapid start from cold is possible.

Comparison between high pressure and conventional boiler:

(1) Fuel burning process: High pressure boilers can be utilized fuels like oil, gas or coal; they have modified burning system and special types of burner.

(2) Flue path: Pressurized combustion provides uniform heating, minimum friction losses, reducing overheating and maximum heat transfer is possible.

(3) Circulation of water: Forced circulation of water provides high heat transfer rate and less scale formation tendency.

(4) Size/shape/space: Compact, no drum, less space requirement.

(5) Design/arrangement: Better design and different parts arrangement lead to increased heat transfer surface area, so increased efficiency of boiler.

(6) Pressure/capacity range: Steam is generated at high pressure 160 bar and above even supercritical pressure (above 225 bar), steaming capacity 30 to 650 tones/hr, temperature range 500–540°C.

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(7) Fuel consumption: Lower as 250 Tones/hr

(8) Waste disposal/draught system: High pressure boiler uses modified draught system and hence, there is reduction in pollutions.

2.3 La-Mont boiler [Dec. '13]

The La-Mont boiler is a first forced circulation boiler introduced by La-Mont in 1925 in which pump is used for forced circulation.

Type of boiler: Water tube, forced circulation, high pressure boiler.

Steam generating capacity: 45 to 50 tones/hr

Pressure and temperature: 120 atm. and 500° C

Working:

The feed water from the hot well is pumped to boiler drum through economizer. The economizer as shown in Fig. 2.4 is located in the boiler where hot combustion gases pass.

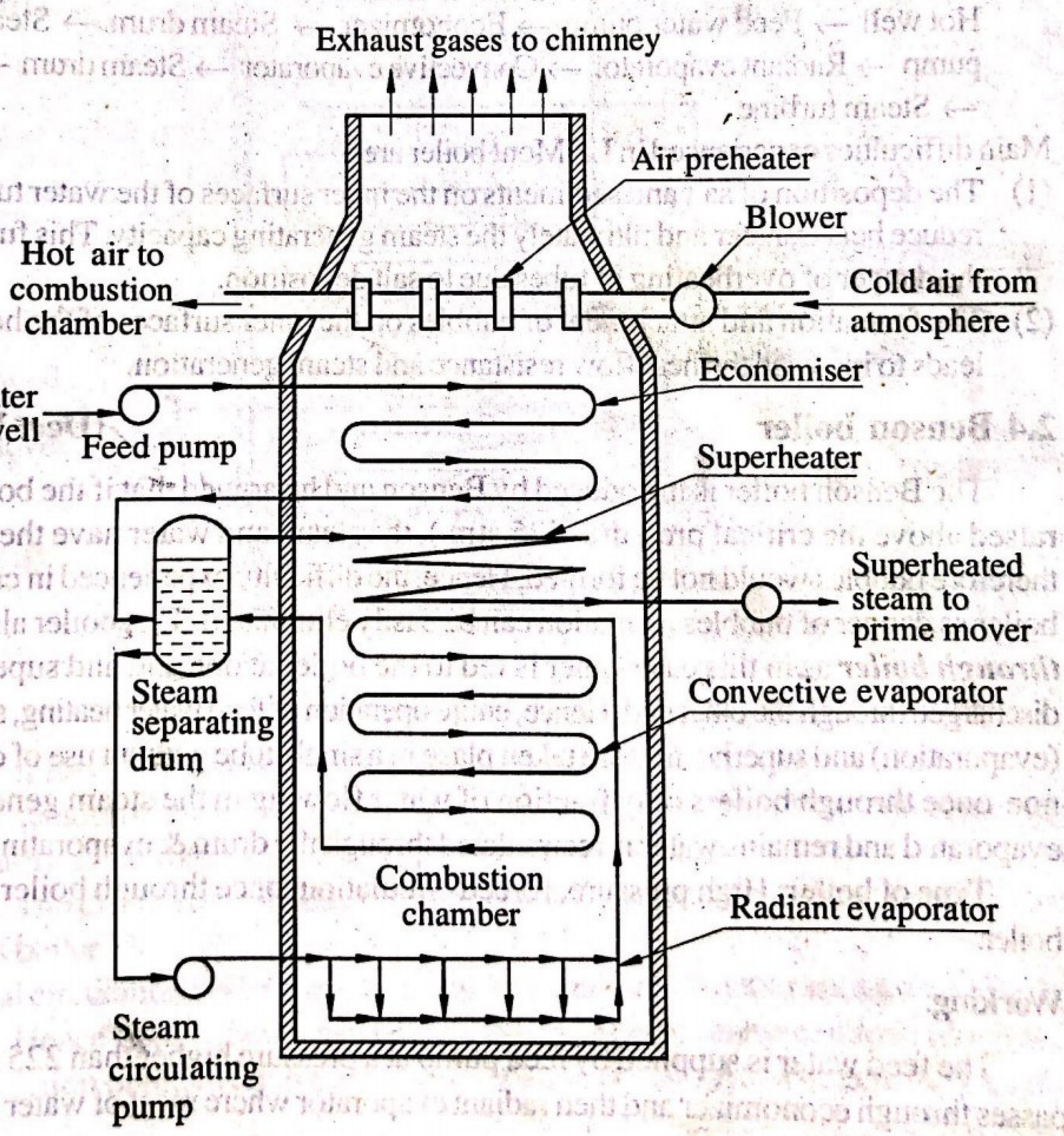


Fig 2.4 La-Mont boiler

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In economizer, the feed water absorbs the sensible heat from flue gases passes over the economizer. The water from the drum is pumped to radiant and convection evaporator, and again to drum, The feed water in radiant evaporator absorb radiant heat and in convective evaporator absorb convective heat and it evaporates.

The evaporated water (steam) collected in the upper part of separating steam drum. Then it passes through superheater where it becomes superheated. The superheated steam is supplied to the prime mover.

The atmospheric cold air is preheated in the air-preheater by the flue gases before they are discharged through the chimney. The heated air from air preheater is supplied to the combustion chamber.

The combustion product (flue gases) flow in following sequence :

Combustion chamber → Convective evaporator → Superheater → Economizer → Air preheater → Chimney → Atmosphere

The water/steam flow in following sequence :

Hot well → Feed water pump → Economizer → Steam drum → Steam circulating pump → Radiant evaporator → Convective evaporator → Steam drum → Superheater → Steam turbine

Main difficulties experienced in La-Mont boiler are :

(1) The deposition of salt and sediments on the inner surfaces of the water tubes. It leads to reduce heat transfer and ultimately the steam generating capacity. This further increases the danger of overheating of tubes due to salt deposition.

(2) The formation and attachment of bubbles on the inner surfaces of the heating tubes. It leads to increase the heat flow resistance and steam generation.

2.4 Benson boiler [Dec. '13, June '11]

The Benson boiler is introduced by Benson and he argued that if the boiler pressure is raised above the critical pressure (225 atm.), the steam and water have the same density; therefore bubbles would not be formed. Hence, the difficulty experienced in case of La-mont boiler as danger of bubbles formation can be easily eliminated. This boiler also known ***once through boiler*** as in this case water is fed to the boiler at one end and superheated steam discharged through the other end. Hence, entire operation of feed water heating, steam formation (evaporation) and superheating are taken place in a single tube without use of drum. In case of non-once through boilers only fraction of water flowing in the steam generating tubes is evaporated and remains water is recirculated through the drum & evaporating tube.

Type of boiler: High pressure, forced circulation, once through boiler and water tube boiler.

Working:

The feed water is supplied by feed pump at a pressure higher than 225 atm. This water passes through economizer and then radiant evaporator where most of water is evaporated as

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shown in Fig. 2.5. The remaining water is evaporated in convective evaporator and converted into steam. The steam is passed through the superheater and finally supplied to the prime mover.

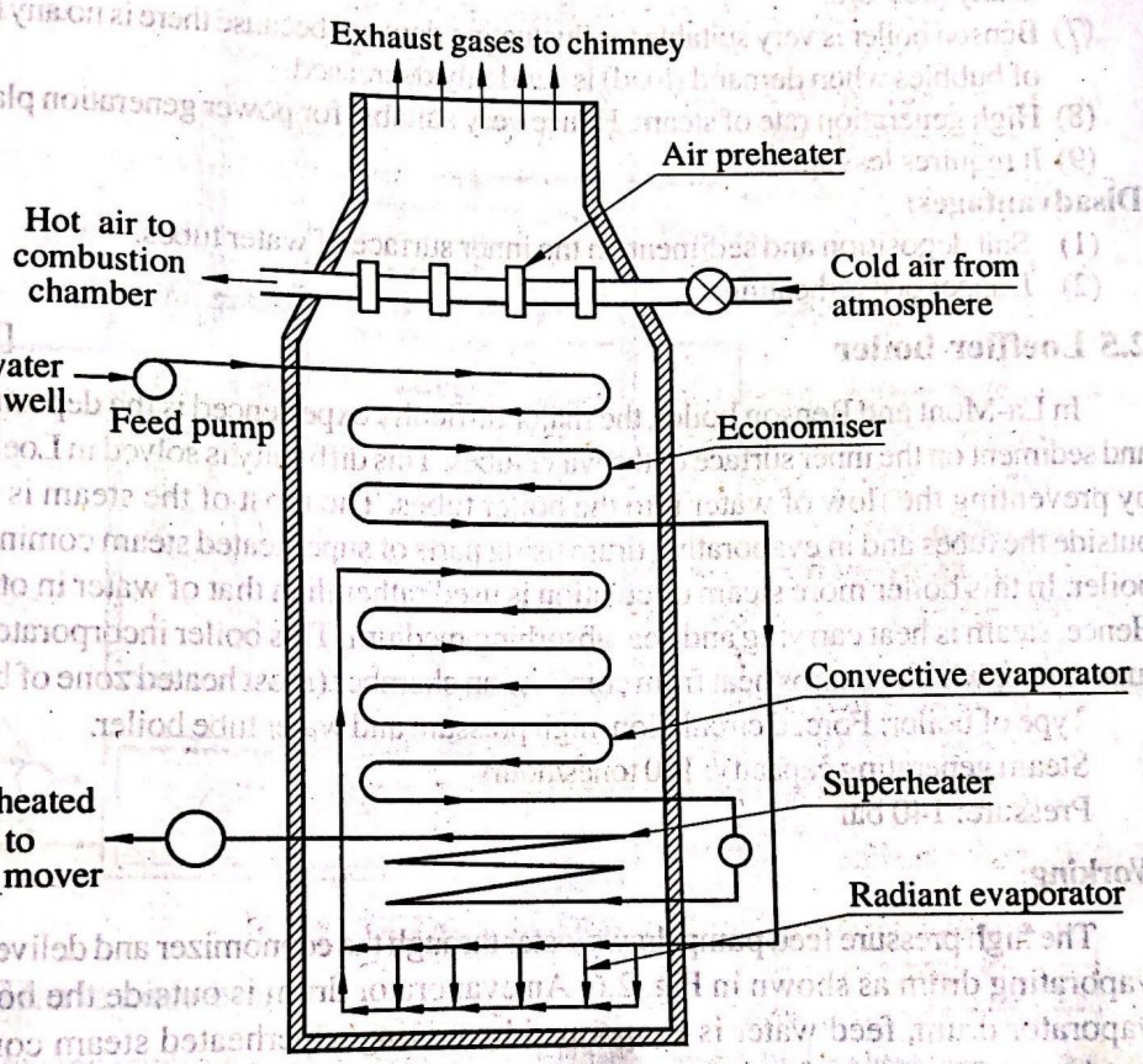


Fig. 2.5 Benson boiler

Advantages of Benson boiler:

(1) The generation of steam in evaporating tubes at pressure higher than critical pressure (225 atm). Therefore, do not require any evaporator drum.

(2) There is no drum; total weight of boiler is 20 % less than other boiler. It reduces the cost of boiler.

(3) Natural circulation boiler requires expansion joints but it does not require in Benson boiler. Hence Benson boiler can be started very quickly because of welded joints.

(4) The erection of the Benson boiler is easier and quicker as all the parts are welded at site.

(5) The superheater of Benson boiler is integral part of the forced circulation and hence no

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special starting arrangement for superheater is required.

(6) At part and overload the Benson boiler can be operated economically by varying temperature and pressure. The desired temperature of steam can be maintained constant at any pressure.

(7) Benson boiler is very suitable for fluctuating demand because there is no any formation of bubbles when demand (load) is suddenly decreased.

(8) High generation rate of steam. Hence very suitable for power generation plant.

(9) It requires less space.

Disadvantages:

(1) Salt deposition and sediment on the inner surface of water tubes.

(2) Danger of overheating.

2.5 Loeffler boiler

[May '13]

In La-Mont and Benson boiler, the major difficulty experienced is the deposition of salt and sediment on the inner surface of the water tubes. This difficulty is solved in Loeffler boiler by preventing the flow of water into the boiler tubes. The most of the steam is generated outside the tubes and in evaporating drum using parts of superheated steam coming from the boiler. In this boiler more steam circulation is used rather than that of water in other boiler. Hence, steam is heat carrying and heat absorbing medium. This boiler incorporates a radiant superheater which absorbs heat from combustion chamber (most heated zone of boiler).

Type of boiler: Forced circulation, high pressure and water tube boiler.

Steam generating capacity: 100 tones/hours

Pressure: 140 bar

Working:

The high pressure feed pump draws water through the economizer and deliver it into the evaporating drum as shown in Fig. 2.6. An evaporator drum is outside the boiler. In the evaporator drum, feed water is evaporated by mixing superheated steam coming from superheater. The steam circulating pump draws saturated steam from the evaporator drum. Then steam passes through radiant and convective superheaters where it is heated to required temperature. Steam coming from superheater, about 1/3 of superheated steam is supplied to steam turbine and the remaining 2/3 superheated steam is supplied to evaporator drum to evaporate the feed water.

Advantages:

(1) The evaporator tubes in this boiler carries superheated steam, therefore there is no salt deposition on the surface of tubes.

(2) Boilers are compact in design.

(3) High steam generating capacity.

(4) This boiler is suitable for marine applications.

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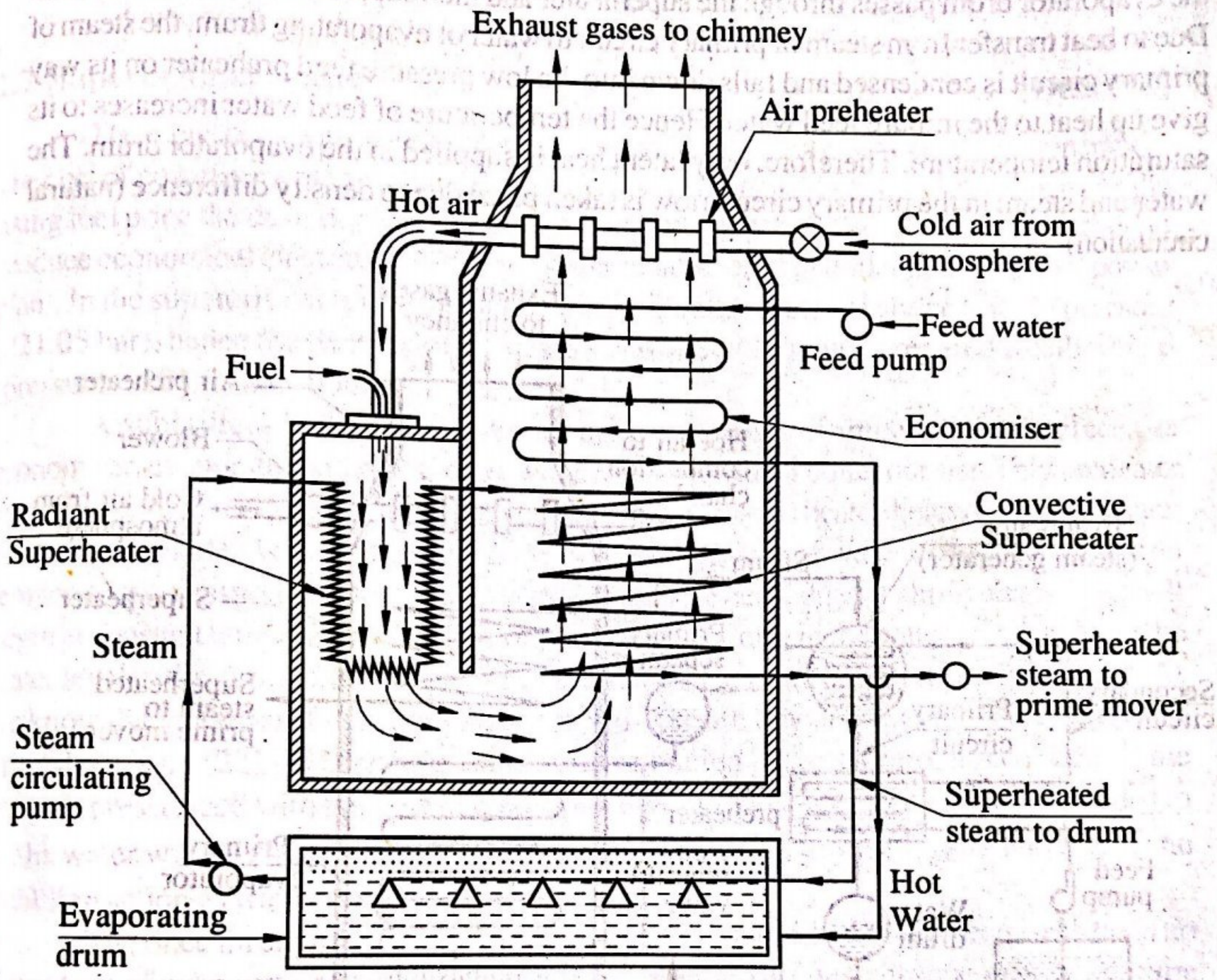


Fig. 2.6 Loeffler boiler

2.6 Schmidt-Hartmann boiler [June '12]

This boiler having multi circuit as primary and secondary circuit which reduces the problem of overheating and salt deposition in the tubes. The operation of boiler is similar to an electric transformer.

In this boiler a feed pump is used to supply water to a drum which discharged saturated steam to the superheater. This main feed water (impure water) is evaporated as pressure about 60 bar in the secondary circuit. The pure distilled water is evaporated at pressure about 100 bar in the primary circuit. In this boiler energy (heat) exchanges between two fluids at different pressures.

In the primary circuit steam is produced from distilled water, this steam passes through a submerged heating coil which is located in an evaporator drum as shown in Fig. 2.7. This high pressure (100 bar) steam in this coil of primary circuit has a sufficient energy (heat) to generate steam from impure water in the evaporator drum at pressure 60 bar. The steam produced in

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the evaporator drum passes through the superheater and then supplied to the steam turbine. Due to heat transfer from steam of primary circuit to water of evaporating drum, the steam of primary circuit is condensed and falls down into the low pressure feed preheater on its way give up heat to the impure feed water. Hence the temperature of feed water increases to its saturation temperature. Therefore, only latent heat is supplied in the evaporator drum. The water and steam in the primary circuit flow is taken place due to density difference (natural circulation).

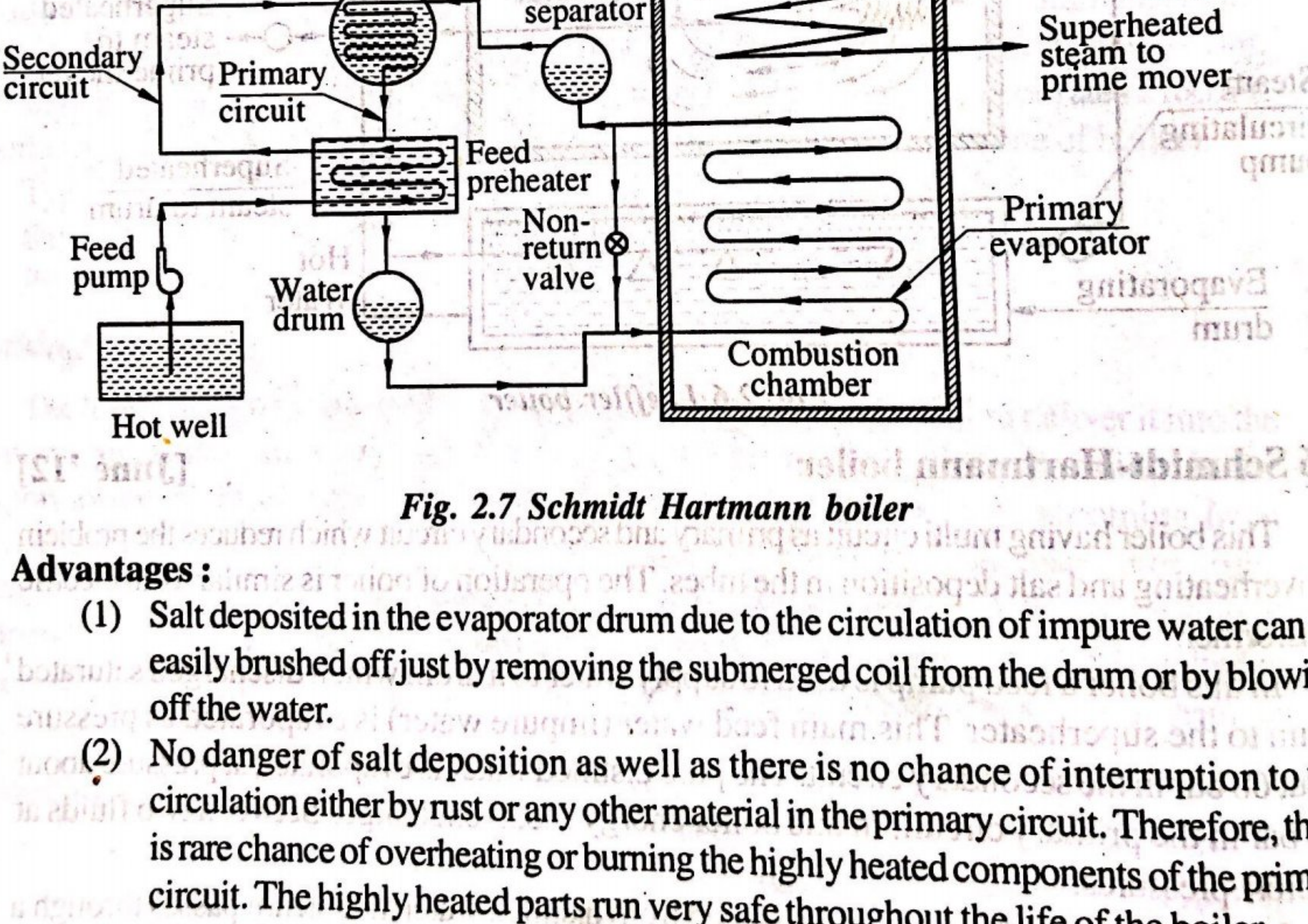


Fig. 2.7 Schmidt Hartmann boiler

Advantages :

(1) Salt deposited in the evaporator drum due to the circulation of impure water can be easily brushed off just by removing the submerged coil from the drum or by blowing off the water.

(2) No danger of salt deposition as well as there is no chance of interruption to the circulation either by rust or any other material in the primary circuit. Therefore, there is rare chance of overheating or burning the highly heated components of the primary circuit. The highly heated parts run very safe throughout the life of the boiler.

(3) The wide fluctuations of load are easily taken by this boiler without undue priming or abnormal increase in the primary pressure due to high thermal and water capacity of the boiler.

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(4) The evaporation of feed water takes place in the evaporator drum without priming due to moderate temperature difference across the heating coil.

2.7 Supercritical boiler [Dec. '11]

The prime duty of power plant designer is to reduce the cost of electric power generation. The cost of power generation increases with the increase in the fuel cost. To compensate the rising fuel price the designers continuously try to find the ways. The most recent method to produce economical electric power is by the use of supercritical boiler in the thermal power plant. In the supercritical boiler the boiler working under pressure above critical pressure (221.05 bar), hence the heat transfer rates are considerably large compared to subcritical (pressure < 221.05 bar) boiler.

A subcritical boiler (conventional boiler) consists of three heating surfaces as economiser, evaporator and superheater while the supercritical boiler consists only preheater (economiser) and superheater. In the conventional boiler water is heated below critical pressure, hence heat added to water is utilized to increase temperature of water up to their saturation temperature corresponding to pressure of boiler. If more heat added to this water boiling will begin at constant temperature. Bubbles of steam begin to form in the water and rise above the water level in the drum. Further heating of saturated steam, steam becomes superheated. As we know that pressure of water or steam is raised, the enthalpy of evaporation is reduced. At critical pressure (221.05 bar) the enthalpy of evaporation becomes zero. It means that if the water is pressurised with the help of feed pump upto critical pressure and then heat added to it, the water will be directly converted to steam at the temperature of 374°C and there is no bubbling action as water changes to steam.

The once through boiler is only type suited to supercritical boiler because there is no latent heat of evaporation involved. When water is heated at constant supercritical pressure suddenly it is converted into steam. As shown in Fig. 2.8, the high pressure (above critical point) water enters the tube inlets and leaves at the outlet as the superheated steam. Since, there is no drum, but there should be a transition section where the water is likely to flash in order to accommodate the large increase in volume.

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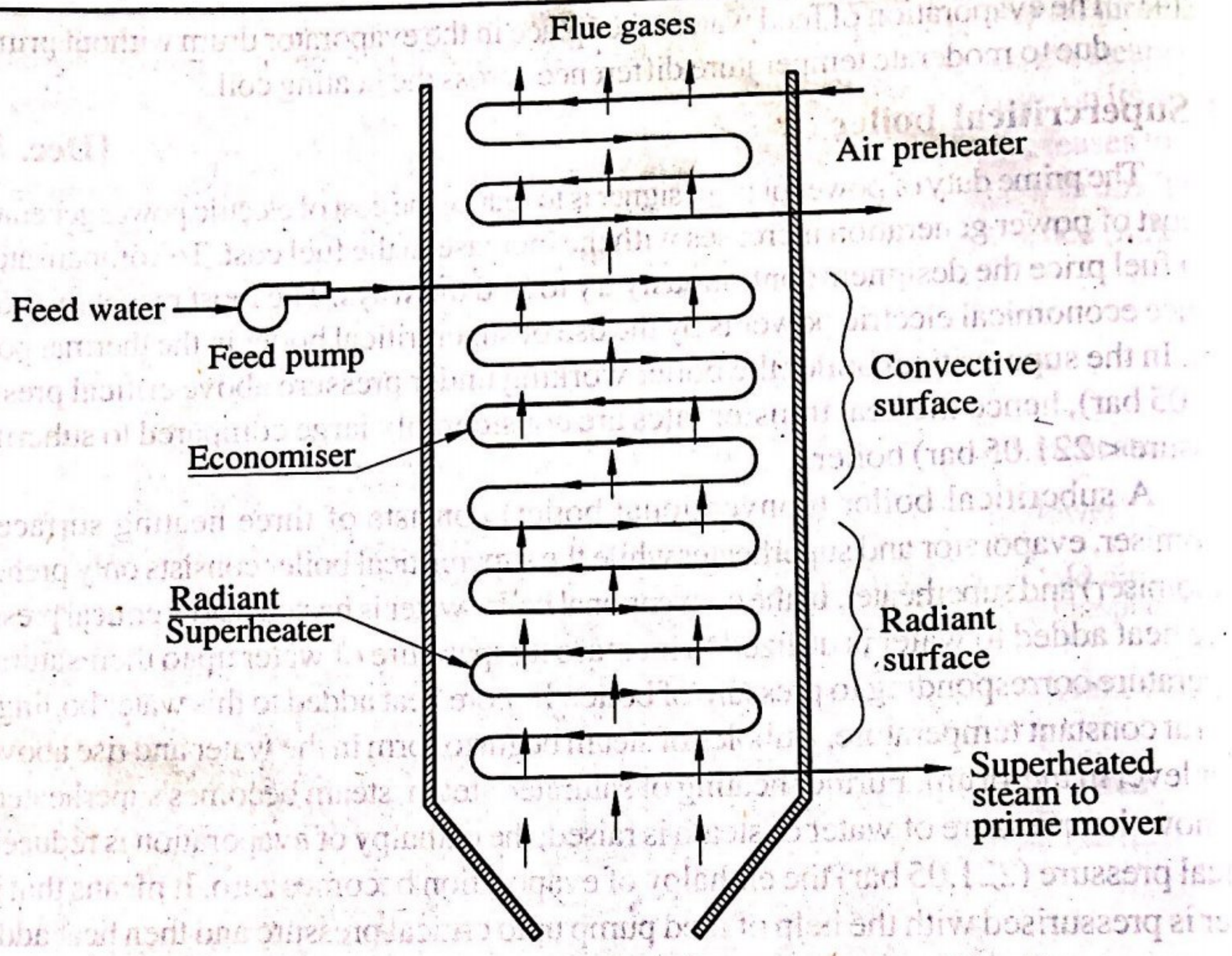


Fig. 2.8 Supercritical boiler

Advantages of supercritical boiler :

(1) Heat transfer rates are considerably large compared to subcritical boilers.
(2) There is no drum, less heat capacity of the generator and hence more stable and gives better response.
(3) There is no two phase mixture and hence the problem of erosion and corrosion are minimized.
(4) The turbo-generator connected to super-critical boilers can generate peak loads by changing the pressure of operation.
(5) There is great ease of operation and their comparative simplicity and flexibility made them adaptable to load fluctuations.
(6) Higher thermal efficiency (about 40 to 42%) of power station can be achieved with use of supercritical boiler.

Disadvantages :

(1) The high pressure and temperature of supercritical boiler restricts use of boiler due to availability of material and difficulties experienced in the turbine and condenser operation due to large volumes.
(2) The additional problem is created due to the separation of solid impurities as phase changes. These solids remain in the tubes and block the passage for the flow of feed

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water. Therefore it is necessary to treat water thoroughly before supply to the boiler.

2.8 Supercharged boiler

As we know that when the combustion is carried out under high pressure in the combustion chamber by supplying the compressed air then rate of heat transfer is increased and heating surfaces required is reduced. This theory is used in supercharged boiler as shown in Fig. 2.9.

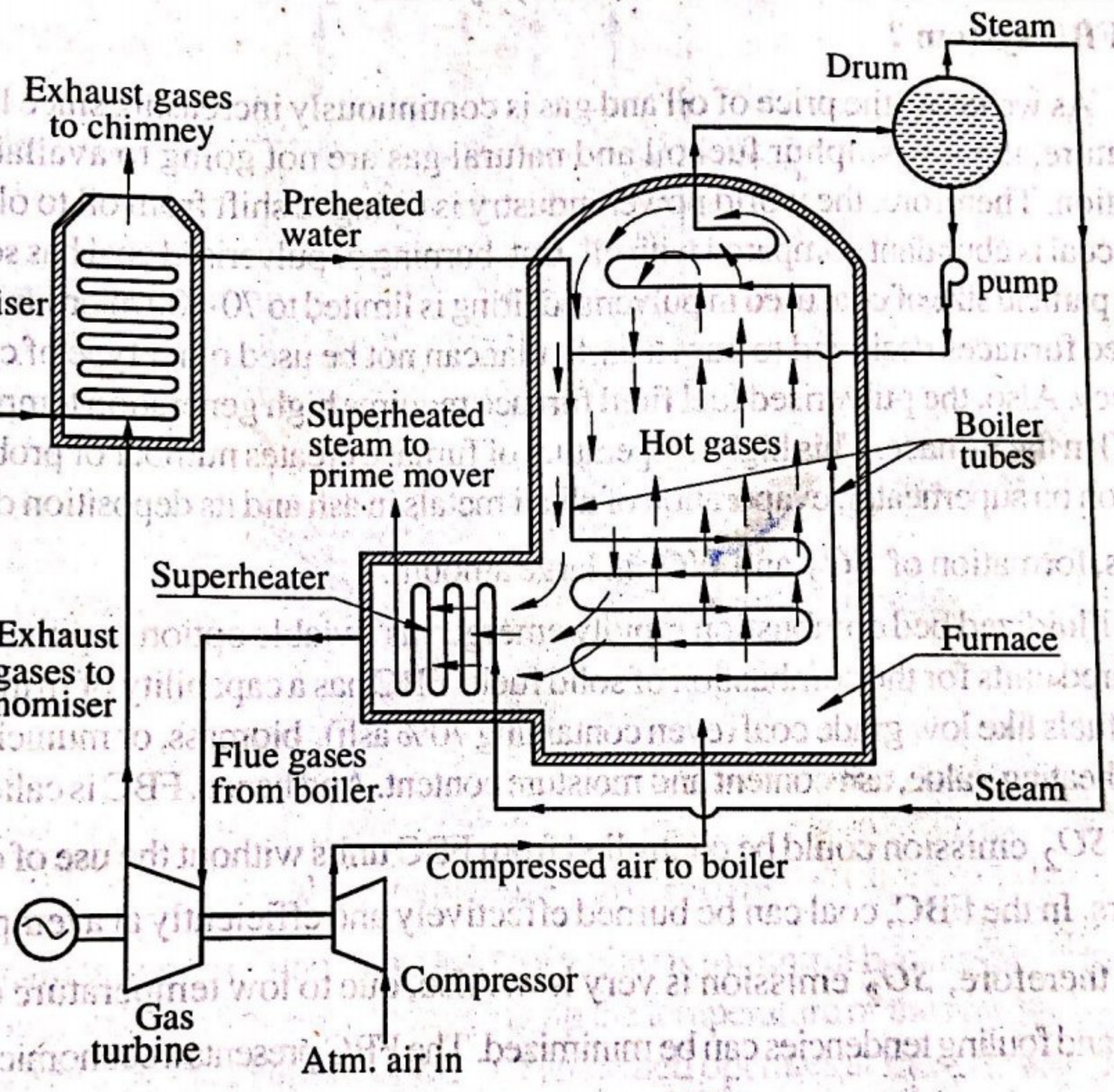


Fig. 2.9 Supercharged boiler

The high pressure compressed air is supplied to combustion chamber. Part of heat of hot gases in the furnace is absorbed by boiler tubes in which water evaporated and then it is superheated in superheater. The high pressure and temperature exhaust gases from combustion chamber are used to run gas turbine. The work produces by gas turbine is used to run air compressor. The exhaust gases coming from gas turbine passes over the economizer tubes. Then escape to the atmosphere through chimney. In the economizer, the feed water heated by high pressure exhaust gases. Usually pressure of gas side is 5 bar and pressure to the steam side of 200 bar are preferred.

Advantages of supercharged boiler :

(1) Supercharged boiler requires 30 to 25% of heat transfer surface of conventional

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boiler due to very high overall heat transfer co-efficient.

(2) Due to small heat capacity of the boiler, boiler plant gives better response to control.

(3) Rapid start of the boiler is possible due to less heating surfaces and compactness.

(4) The part of the gas turbine output can be used to drive other auxiliaries.

(5) Comparatively less number of operators are required.

Disadvantages : It requires tight passage for high pressure gas.

2.9 Fluidized Bed Combustion (FBC) system [June '12]

Why FBC system ?

As we know the price of oil and gas is continuously increasing since last decade. In near future, the low sulphur fuel oil and natural gas are not going to available for power generation. Therefore, the world power industry is trying to shift from oil to old faithful fuel coal as coal is abundant compared with oil. But, burning of pulverised coal has some problems such as particle size of coal used in pulverised firing is limited to 70-100 microns and pulverised fuel fired furnaces designed to burn a particular can not be used other type of coal with same efficiency. Also, the pulverised fuel fired furnace requires high generation temperature (about 1650°C) in the furnace. This high temperature of furnace creates number of problems like slag formation on superheater, evaporation of alkali metals in ash and its deposition on heat transfer surfaces, formation of SO_2 and NO_2 in large amount.

Fluidized bed combustion rapidly emerged as a viable option to pulverized coal and stoker fired units for the combustion of solid fuels. FBC has a capability of firing a wide range of solid fuels like low grade coal (even containing 70% ash), biomass, or municipal waste with varying heating value, ash content and moisture content. And hence, FBC is called fuel flexible system. SO_2 emission could be controlled from FBC units without the use of costly external scrubbers. In the FBC, coal can be burned effectively and efficiently at a temperature below 1100°C, therefore, SO_2 emission is very low. Also, due to low temperature of furnace, the slagging and fouling tendencies can be minimized. The FBC presents a economical and practical way to burn low quality coal with minimum SO_2 and NO_2 emission.

Principle of FBC System :

A fluidized bed is composed of fuel and inert material contained with an atmospheric or pressurised vessel. When air or other gas flows upward through bed, the bed solid particles are disturbed. If velocity increased further a stage is reached when the particles are suspended in the gas stream and the composed (packed) bed becomes turbulent and rapid mixing of particles occurs. The behavior of this mixture of solid particles and air or gas is like a fluid. Burning of a fuel in such a stage is known as ***fluidized bed combustion***.

Fig. 2.10 shows principle of FBC system, the mixture of fuel (crushed coal) and inert material (crushed dolomite or limestone) are fed on a distribution plate and air is supplied from the bottom of distribution plate. The air is supplied at high velocity so that solid particles

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of feed material remains in suspension condition during burning. The velocity of air causing fluidization is dependent on a given solid particle size, range and density. The heat produced is used to heat water flowing through evaporator tube and convert water into steam. The evaporator tubes are directly immersed in the fluidized bed and direct contact between the burning coal particles and tubes produce very high heat transfer rates. This reduces the size of the unit.

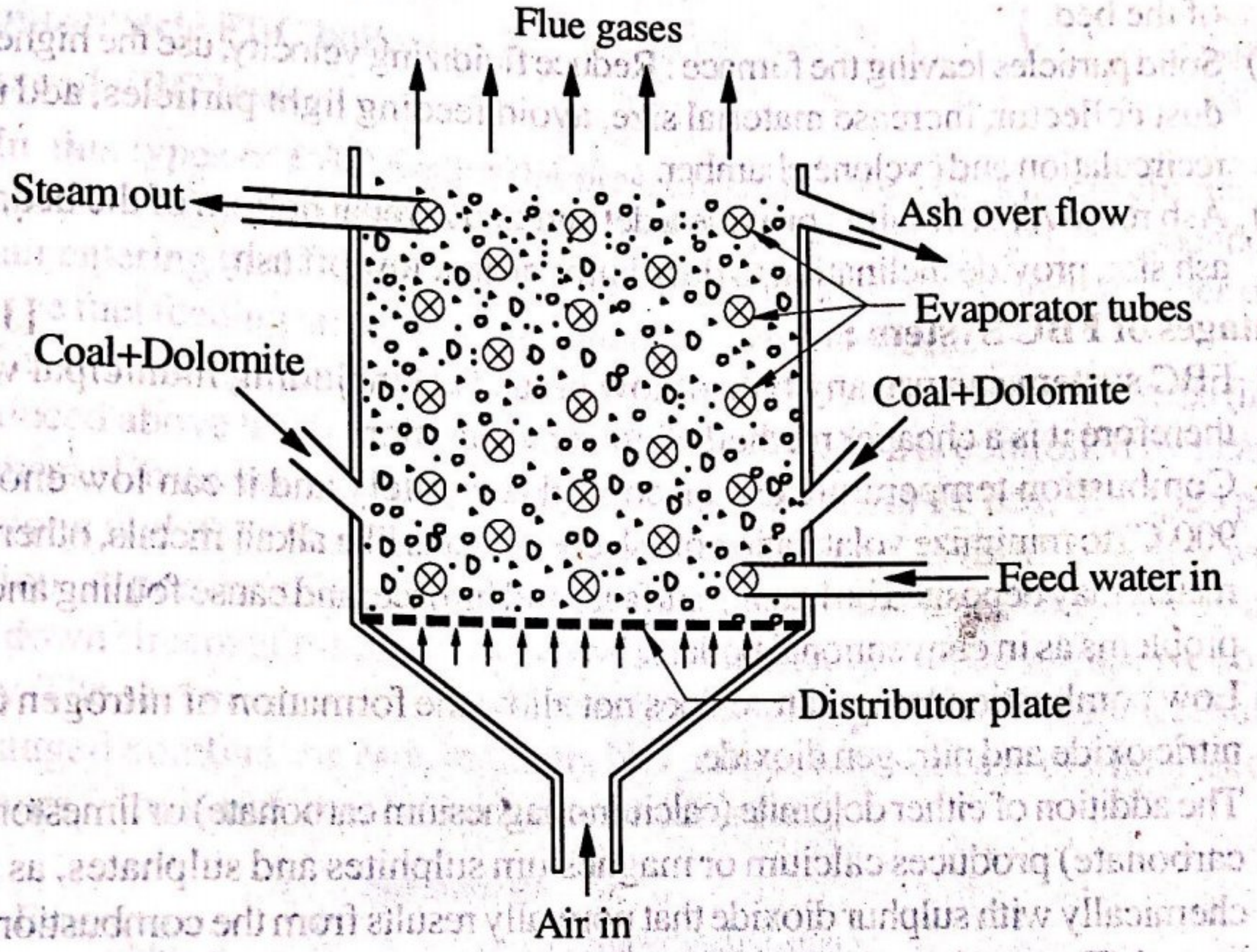


Fig. 2.10 Basic FBC system

The main function of inert material (dolomite) is to control bed temperature. The heat released by combustion is first used in keeping up the temperature of the inert material and the balance is absorbed by the evaporator tubes. Hence bed operates at much lower temperature about 800-900°C. Another function of inert material (dolomite) is to absorb the SO_2 and bring down SO_2 emmision level to 15% of that in conventional firing methods. The amount of NO_x produced is also reduced because of low temperature of bed and low excess air as compared to pulverized fuel firing. The inert material should be resistant to heat and disintegration and should have similar density as that of coal. Lime stone, dolomite, fused alumia, sintered ash are commonly used as inert materials.

Main problems or weaknesses with conventional FBC

(1) Short residence time of fuel in bed.
(2) Poor lateral mixing of air and fuel.
(3) Fine solid particles leaving the furnace along with the flue gases.
(4) Very difficult to remove ash and incombustible.

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Treatment to overcome the above mentioned difficulties :

(1) Short residence time : Fuel and inert material should be fine, recirculate the unburnt carbon, supply more inert material than optimum amount, reduce the velocity of bed, avoid feeding light solids and liquids.

(2) Poor lateral mixing of air and fuel : Use more air, use higher calcium-sulphur ratio, fuel introduce at more points, avoid using smoky and smelly fuels as this causes clinkering of the bed.

(3) Solid particles leaving the furnace : Reduce fluidizing velocity, use the higher capacity dust collector, increase material size, avoid feeding light particles, add recapture, recirculation and cyclone chamber.

(4) Ash removal difficulty : provide a defluidized layer at bottom of the bed, introduce ash slot, provide inclination to distributor plate, blow off ash.

[Jan. 2013]

Advantages of FBC System :

(1) FBC system can use any type of low grade fuel including municipal wastes and therefore it is a cheaper method of power generation.

(2) Combustion temperature can be control accurately and it can low enough (800-900°C) to minimize volatization of ash constituents like alkali metals, otherwise alkali metals may deposited on heating surfaces and furnace, and cause fouling and corrosion problems as in conventional boilers.

(3) Low combustion temperature does not allow the formation of nitrogen oxides like nitric oxide and nitrogen dioxide.

(4) The addition of either dolomite (calcium-magnesium carbonate) or limestone (calcium carbonate) produces calcium or magnesium sulphites and sulphates, as these react chemically with sulphur dioxide that normally results from the combustion of sulphur in coal. These salts are solids and remain in the combustion chamber and they don't go into the atmosphere through the chimney. Thus pollution is controlled and combustion of high sulphur coal is possible.

(5) The size of coal does not effect on the operation and performance of FBC system. The particle size preferable is 6 to 13 mm but even 50 mm size coal can also be used in this system. Hence the cost of coal crushing is reduced.

(6) FBC system can respond rapidly to changes in load demand since thermal equilibrium between air and coal particles in the bed is quickly established.

(7) Due to better heat transfer, higher thermal efficiencies, the unit size and hence the capital costs are reduced. No moving parts, simple ash and coal handling system, less excess air required (smaller fan size), hence less operating and maintenance costs. There is a saving of about 10% in operating cost and 15% in the capital cost of power plant.

Disadvantages :

[Jan. 2013]

(1) There may be problems in distribution of coal and dolomite.

(2) The direct mechanical contact between hot fuel and evaporator tubes causes erosion

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of the tubes.

(3) High pressure air required to fluidization of the bed.

Classification of FBC boilers or systems :

1. Atmospheric FBC boilers
   (i) Classical FBC boiler - overfeed or underfeed (ii) Circulation FBC boiler
2. Pressurized FBC boilers.

2.9.1 Atmospheric FBC boilers

(i) Classical FBC boiler

In this types of FBC boiler, the pressure inside the bed is atmospheric. The bed consisting about 97% limestone or inert material and 3% burning fuel, is suspended by hot primary air entering the bottom of the combustion chamber. There are two types of system based on the fuel feeding arrangement as underfeed and overfeed.

The overfeed system is shown in Fig. 2.11 in which the fuel and inert material (limestone) are introduced above the fluidized bed. The bed temperature is controlled by heat transfer tubes immersed in the bed and by varying the quantity of coal in the bed. The solid particles is leaving along with flue gases as coal particle size decreases as result of either wearing or combustion. A portion of the solid particles left from the bed are collected by a cyclone collector down stream of the convection pass and returned to the bed to improve combustion efficiency. The secondary air is added above the bed to improve combustion efficiency and achieve staged combustion thus reducing NO_x emmisions. In case of underfeed fuel and limestone are introduced from bottom of the fluidized bed.

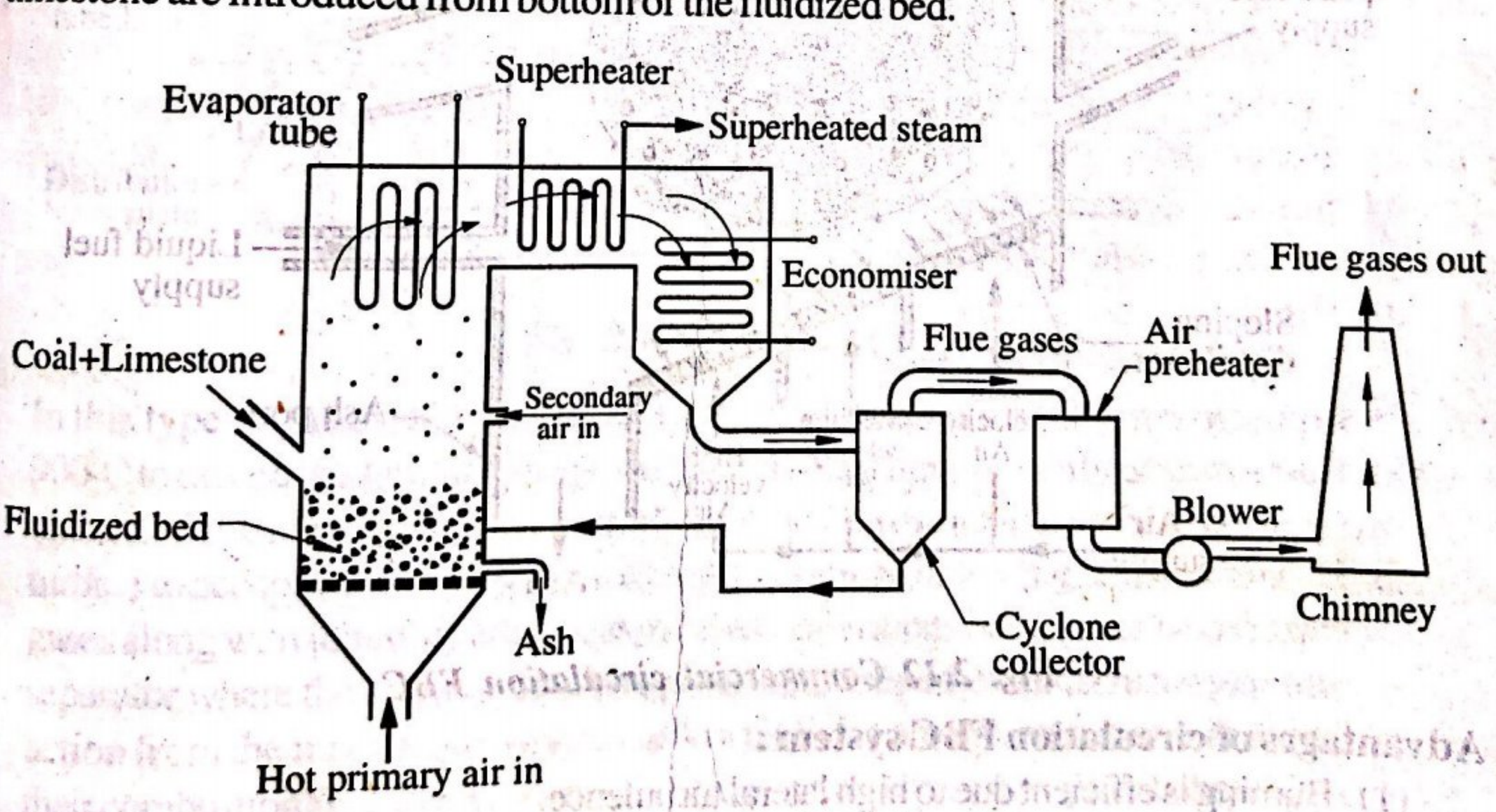


Fig. 2.11 Classical FBC system

The overfeed system is simple in operation and economical in running but results in smaller output per m^2 area and gives poor desulphurization performance. Under feed system provides positive load and a compact design but costly in operation.

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The main disadvantage using the classical FBC is that incombustible particles (ash and metals) of fuel are came downwards and block the distributor when light materials like wood dust, agricultural waste etc. are used as a fuel.

(ii) Circulation FBC boiler

The main disadvantage of classical FBC is to block the distributor. To overcome this difficulty, a group of engineers during 1968-70 suggested the modifications to the simple FBC as (a) provide nonuniform fluidizing velocities over the bed, (b) provide slope to one of the FBC walls over the bed to contain light particles, and (c) provide sloping distributor plate to give an air slide particles. Due to these suggestions there is significant improvement and allowed to use light materials as fuels most successfully. Light materials were burned within the bed and heavy incombustibles (ash and metals) gathered at the bottom of the sloping distributor.

Fig. 2.12 shows commercial circulation FBC system. The solid fuel enters the furnace from the side of walls. The low velocity (LV), medium velocity (MV) and high velocity (HV) air supplied at different points along the sloping surface of the distributor plate. The secondary air is supplied over the bed. The ash port is provided at lower end of the distributor plate.

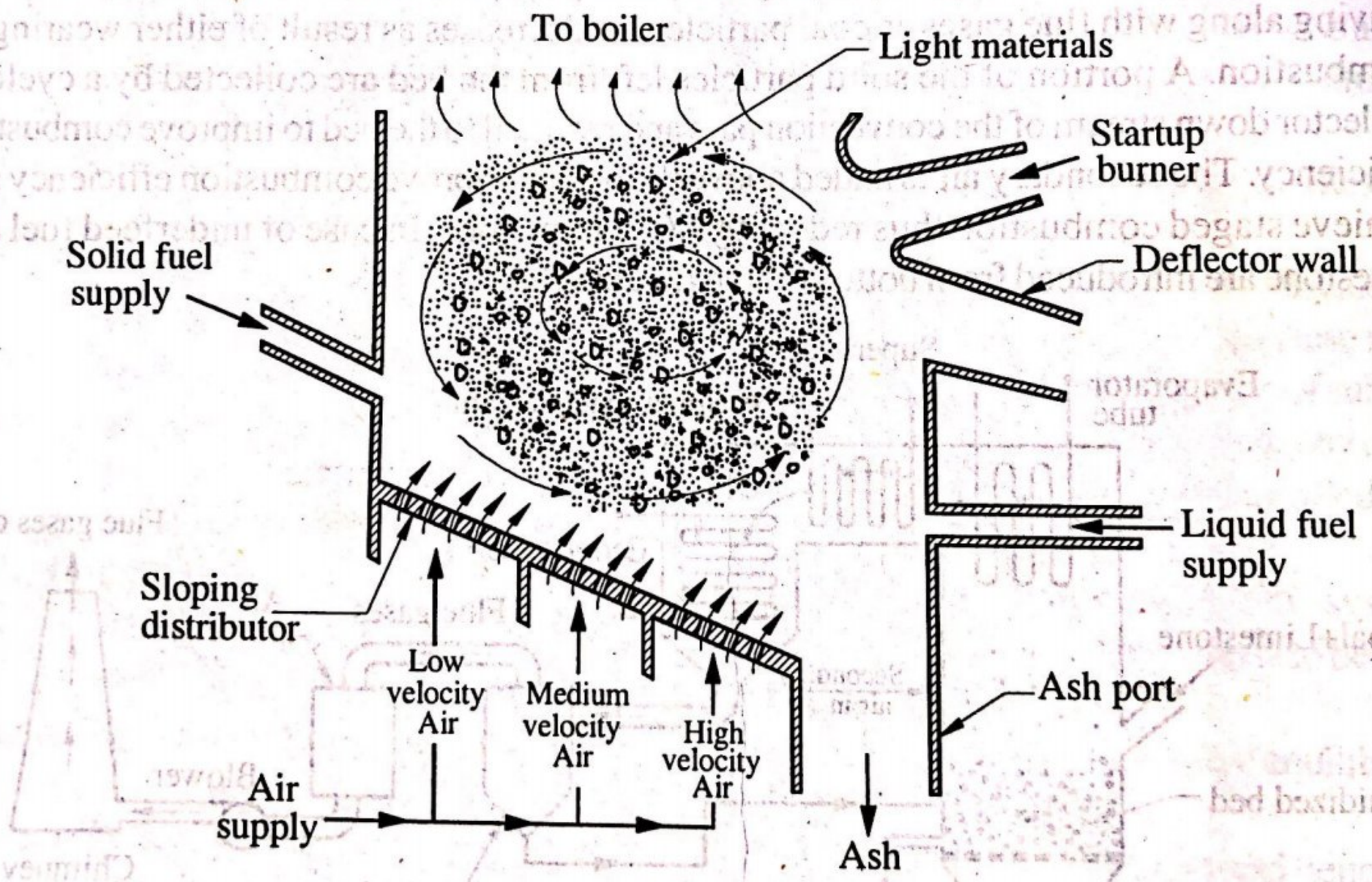


Fig. 2.12 Commercial circulation FBC

Advantages of circulation FBC system :

(1) Burning is efficient due to high lateral turbulence.
(2) Very less solid particle left the furnace along with flue gases.
(3) Due to higher combustion velocities greater fuel enters per unit cross-sectional area is possible.

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(4) Higher combustion efficiency is possible.
(5) Lower NO_x emissions are possible.
(6) Easy removal of ash and uncombustible.
(7) No need of spreader stroker, under bed feeds.
(8) The materials which can be burned are PVC, Wax cartons, waste papers, wood dust, polystyrene, polyamides, tyre rubber, agricultural waste, coals or oils etc.

2.9.2 Pressurized FBC boiler

[Dec. '11]

In this system pressurized air is used for fluidization and combustion. The air is supplied under pressure 10 bar approximately with help of an air compressor as shown in Fig. 2.13.

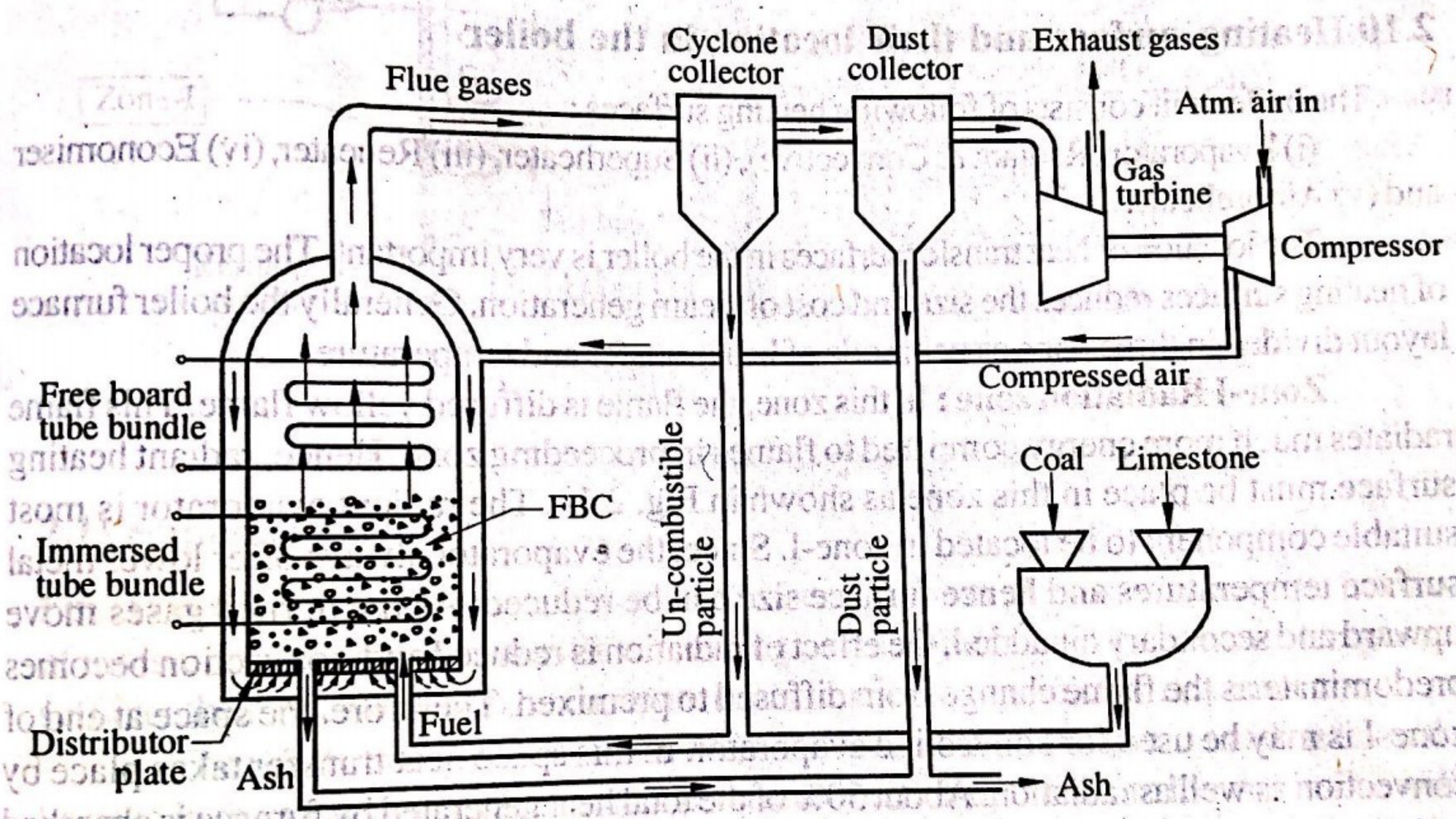


Fig. 2.13 Pressurized FBC boiler

In this type of the system, boiler exit gas contain enough energy about temperature 850 to 900°C to drive a gas turbine. The power output of gas turbine is utilized to run the air compressor and the electric generator. The product of combustion have to be sufficient clean for gas turbine to accept without excessive erosion, corrosion or fouling of the turbine. Hence, the flue gases along with unburnt carbon leaving the combustion chamber are passed through a cyclone separator where the unburned carbon particles are superheated from the gas due to centrifugal action from the mixture, separated carbon particles are returned to fluidized bed to complete their combustion. The flue gases from cyclone separator are now passed through a dust collector. This combined operation of gas turbine with the boiler steam turbine provides an efficient combined cycle operation.

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Advantages of pressurized FBC boiler :

(1) High rate of coal loading is possible.
(2) High rate of burning is achieved.
(3) Comparatively less volume of furnace is required, hence plant size is reduced.
(4) Improved desulphurization and low NOx emission.
(5) High steam generation rate is possible.
(6) Considerable reduction in cost.

Disadvantages :

(1) It is difficult to control.
(2) Plant life is low.

2.10 Heating surface and their location in the boiler

The boiler unit consists of following heating surfaces :

(i) Evaporator (Radiant & Convective), (ii) Superheater, (iii) Reheater, (iv) Economiser and (v) Air preheater.

The location of heat transfer surfaces in the boiler is very important. The proper location of heating surfaces reduces the size and cost of steam generation. Generally the boiler furnace layout divided in three zone as per mode of heat transfer and temperature.

Zone-I Radiation zone : In this zone, the flame is diffused yellow flame. This flame radiates much more energy compared to flames in proceeding zone. Hence, radiant heating surface must be place in this zone as shown in Fig. 2.14. The radiant evaporator is most suitable component to be located in zone-I. Since, the evaporator always offer lower metal surface temperatures and hence furnace size can be reduced. As the burned gases move upward and secondary air added, the effect of radiation is reduced and convection becomes predominate as the flame change from diffused to premixed. Therefore, the space at end of zone-I is may be used for convective evaporator. In this space heat transfer takes place by convection as well as radiation. About 50% of the total heat generated by furnace is absorbed in the radiation zone-I.

Zone II High temperature convection zone : In this zone, heat transfer takes place mainly by convection. This zone is high temperature zone. Superheater elements are more expensive than evaporator because of their high metal surface temperature. It is desirable to locate the superheater in zone-II to reduce its total surface area requirement. Hence, high temperature convection zone (zone II) is highly preferable to locate the superheater and reheater. About 40 to 45% of heat generated by furnace is absorbed in this zone.

Zone-III Low temperature convection zone : In this zone, also heat transfer takes place by convection. The gas temperature in this zone is relatively low, hence zone-III is more appropriate and economical for locating the heat recovery units like economizer and air-preheater. About 4 to 10% of heat generated by furnace is absorbed in zone-III.

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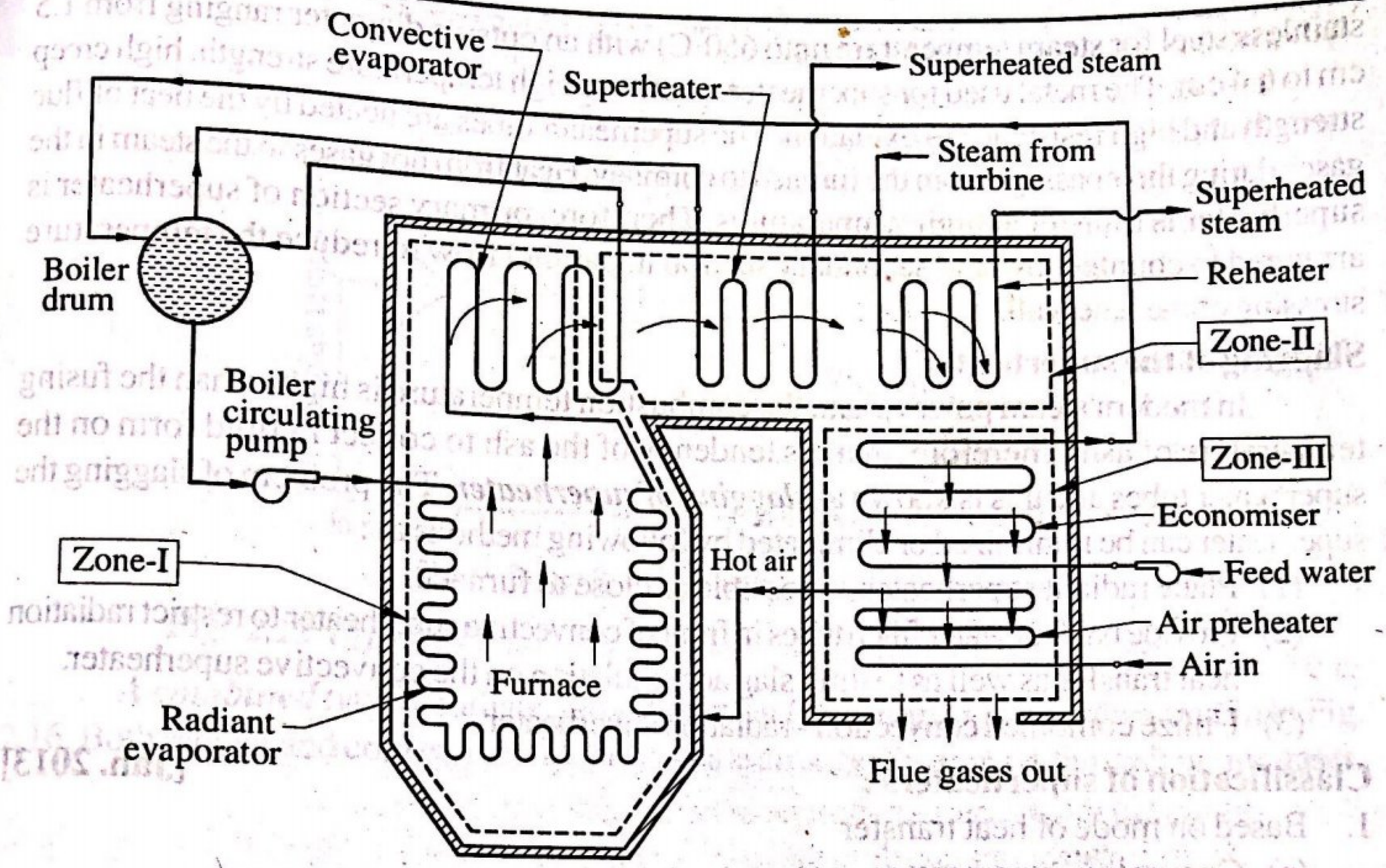


Fig. 2.14 Heating surface of boiler

2.11 Superheaters

Function : The function of superheater in the thermal power plant is to remove the last traces of moisture from the saturated steam coming out of boiler and to increase its temperature sufficiently above saturation temperature as per requirement at inlet of steam turbine.

The steam produced in the boiler is nearly saturated. This steam as such as should not be used in the steam turbine because the dryness fraction of the steam leaving boiler will be low. This results in the presence of moisture which causes corrosion and erosion of turbine blades.

Advantages of superheated steam :

(1) The steam turbine efficiency increased by approximately 2% of each percent of moisture removed and superheating reduces the percentage cut off in steam turbine, therefore steam consumption per unit power output is reduced.

(2) The superheating increases overall cycle efficiency as well as avoids to much condensation in the last stages of turbine which reduces the blade erosion and corrosion.

(3) Superheated steam has a higher heat content and hence capacity to work is increased. Hence an economy in steam consumption in steam turbine is achieved.

Superheaters consists of groups of tubes made of steel (carbon steel for steam temperature upto 510°C, carbon molybdenum steel for steam temperature upto 570°C, and

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stainless steel for steam temperature upto 650°C) with an outside diameter ranging from 1.5 cm to 6.4 cm. The metal used for superheater must have high temperature strength, high creep strength and high resistance to oxidation. The superheater tubes are heated by the heat of flue gases during their passage from the furnace to chimney. Heat from hot gases to the steam in the superheater is transfer at high temperatures. Therefore, primary section of superheater is arranged in counterflow and secondary section in parallel flow to reduce the temperature stressing of the tube wall.

Slagging of the superheater :

In modern steam power plant, the combustion temperature is higher than the fusing temperature of ash. Therefore, there is tendency of the ash to collect in fluid form on the superheater tubes and this is known as ***slagging of superheater***. The problem of slagging the superheater can be minimized or eliminated by following methods as :

(1) Place radiant superheater as possible as close to furnace.
(2) Provide bank of water filled tubes in front of convective superheater to restrict radiation heat transfer as well as to limit slag accumulation on the convective superheater.
(3) Utilize combined convection - radiation superheater.

[Jan. 2013]

Classification of superheaters :

1. Based on mode of heat transfer
   (a) Convective superheater
   (b) Radiant superheater
   (c) Combined convective and radiant superheater (Pendant superheater)
2. Based on their position in the furnace with respect to water tubes
   (a) Overdeck superheater
   (b) Innerdeck superheater
   (c) Inner tube superheater
   (d) Inner bank superheater

Convective superheaters **[June, 2011]** are located in the convective zone of the furnace usually before economizer. The convective superheater absorbs heat from flue gases through convection. The convective superheater gives rising characteristics i.e. the temperature of superheat increases with the increase in steam output as shown in Fig. 2.15 because with the increase the steam output both gas flow over superheater tubes and steam flow inside the tubes increase which causes increase in heat transfer rate and mean temperature difference.

Radiant superheater **[June, 2011]** is located in the furnace between the furnace walls (radiant zone). It absorbs heat from burning fuel through radiation. The radiant superheater has two main disadvantages as it may get overheated due to high temperature of furnace and it has a droping characteristics i.e. the temperature of superheat falls with the increase in steam output as shown in Fig. 2.15, because with the increase in steam output, the furnace temperature rises at a much less rate than the steam output. Since radiant heat transfer is function of furnace temperature the heat transfer increases slowly with steam flow or the steam temperature decreases.

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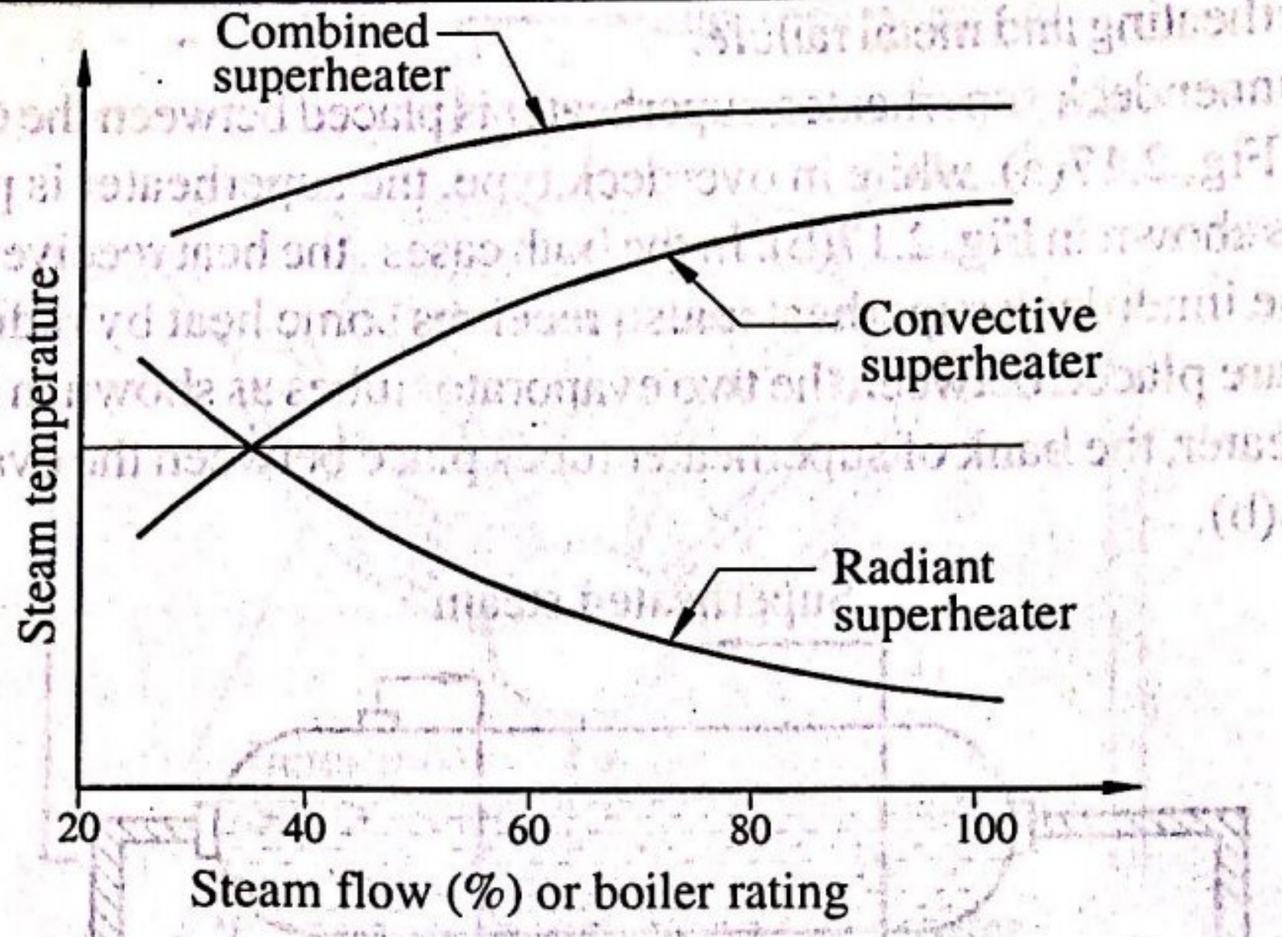


Fig. 2.15 Effect of load (steam output) on superheater temperature

A combined type of radiant and convection (***Pendant***) superheater is shown in Fig. 2.16. Both radiant and convective superheaters are arranged in series in the path of flue gases.

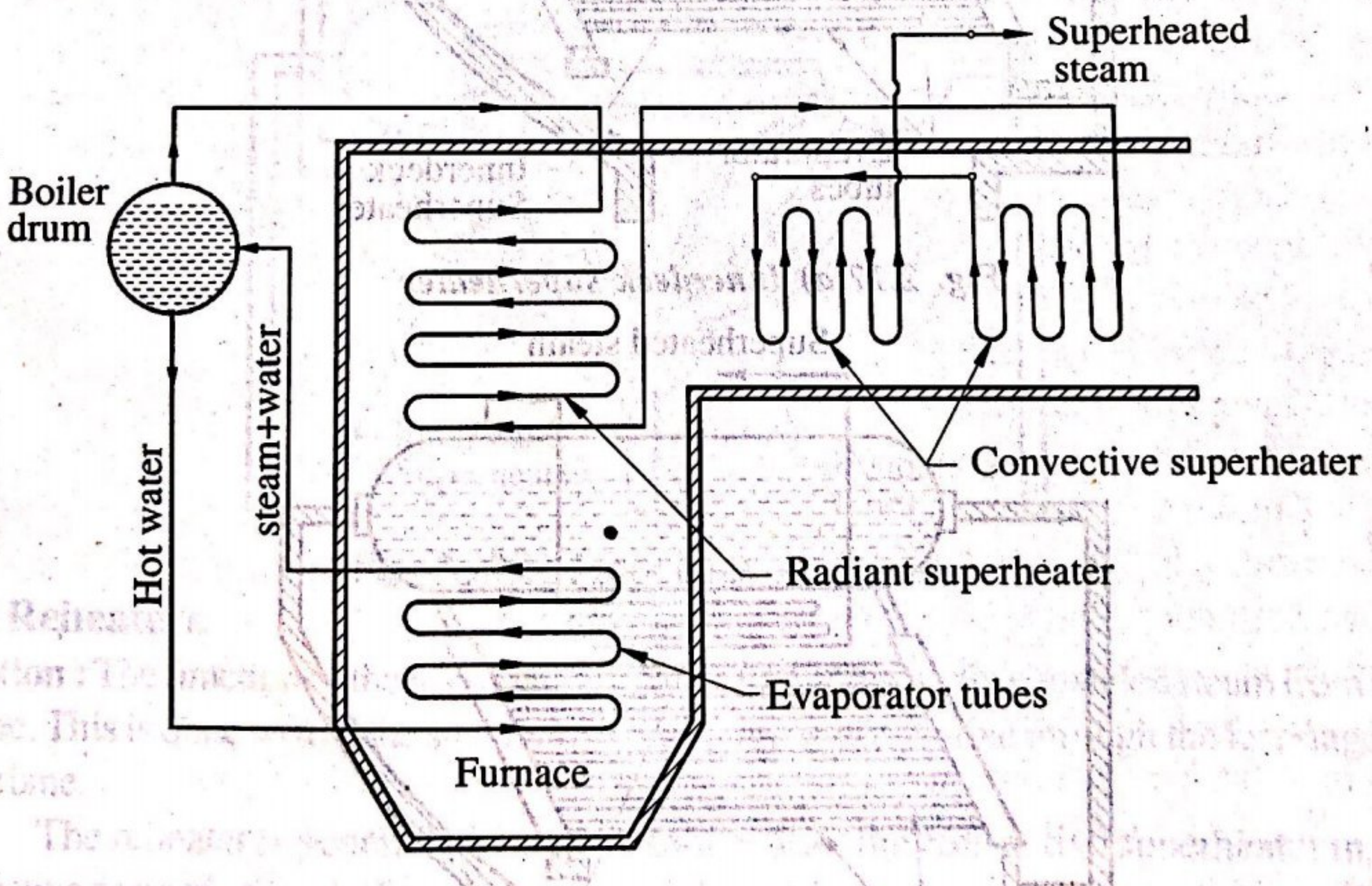


Fig. 2.16 Combined or pendant superheater

Radiant superheater receives heat from the flue gases by radiation and convective superheater receives heat by convection. About 3/4th of convective surface is made counter flow while the remaining 1/4th is parallel flow. If the entire coils are counterflow then exit end of the superheater coil will be subjected maximum temperature difference (outside maximum temperature of gas and inside maximum temperature of steam). Hence, parallel flow exit end of superheater

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protect against overheating and metal failure.

In case of inner deck superheater, superheater is placed between the evaporator water tubes as shown in Fig. 2.17(a), while in overdeck type, the superheater is placed above the evaporator tubes as shown in Fig. 2.17(b). In the both cases , the heat receives the superheater by convection. The innerdeck superheater also receives some heat by radiation. The inner tube superheaters are placed between the two evaporator tubes as shown in Fig. 2.18(a), and inner bank superheater, the bank of superheater tubes place between the evaporator tubes as shown in Fig. 2.18(b).

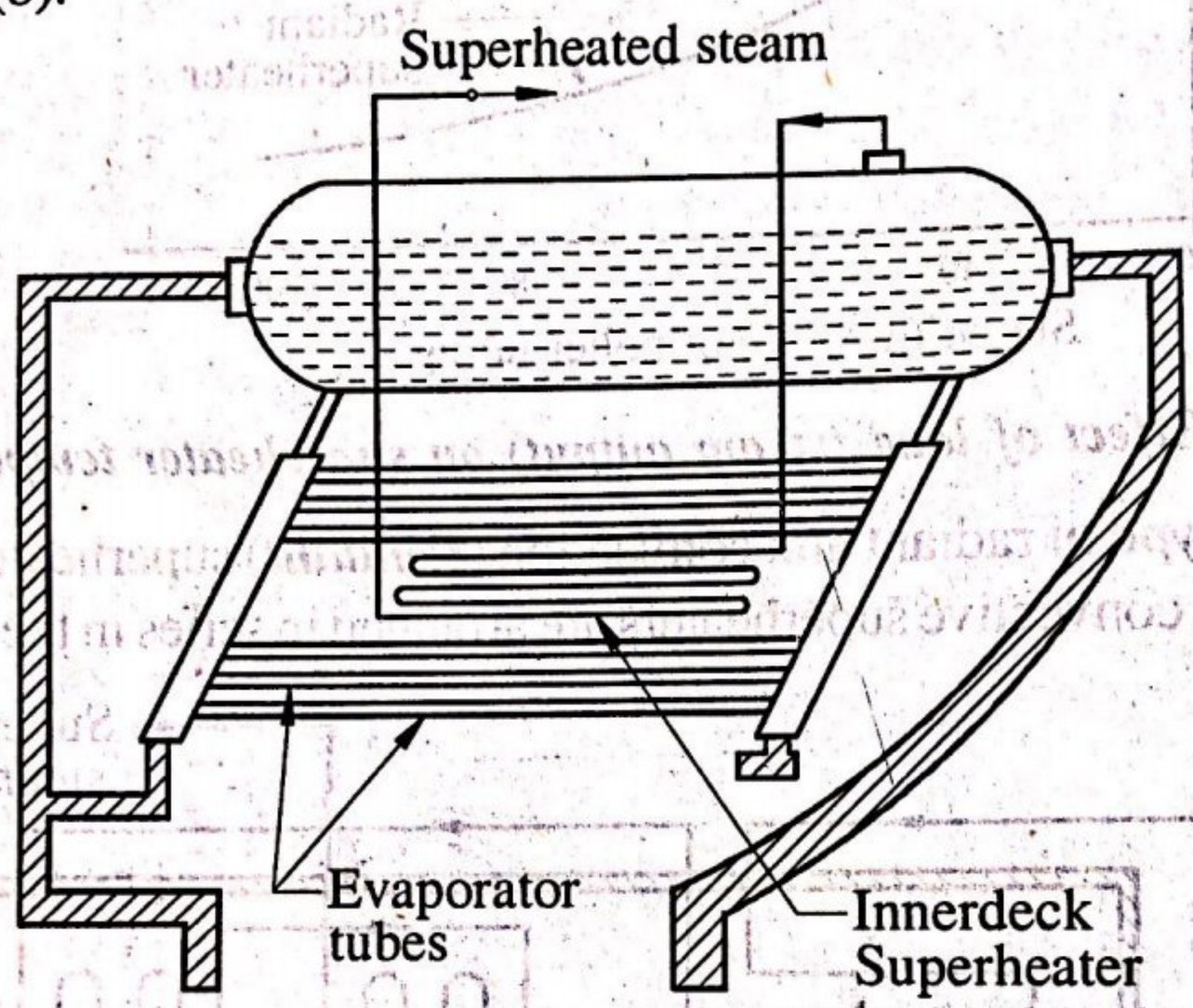


Fig. 2.17(a) Innerdeck superheater

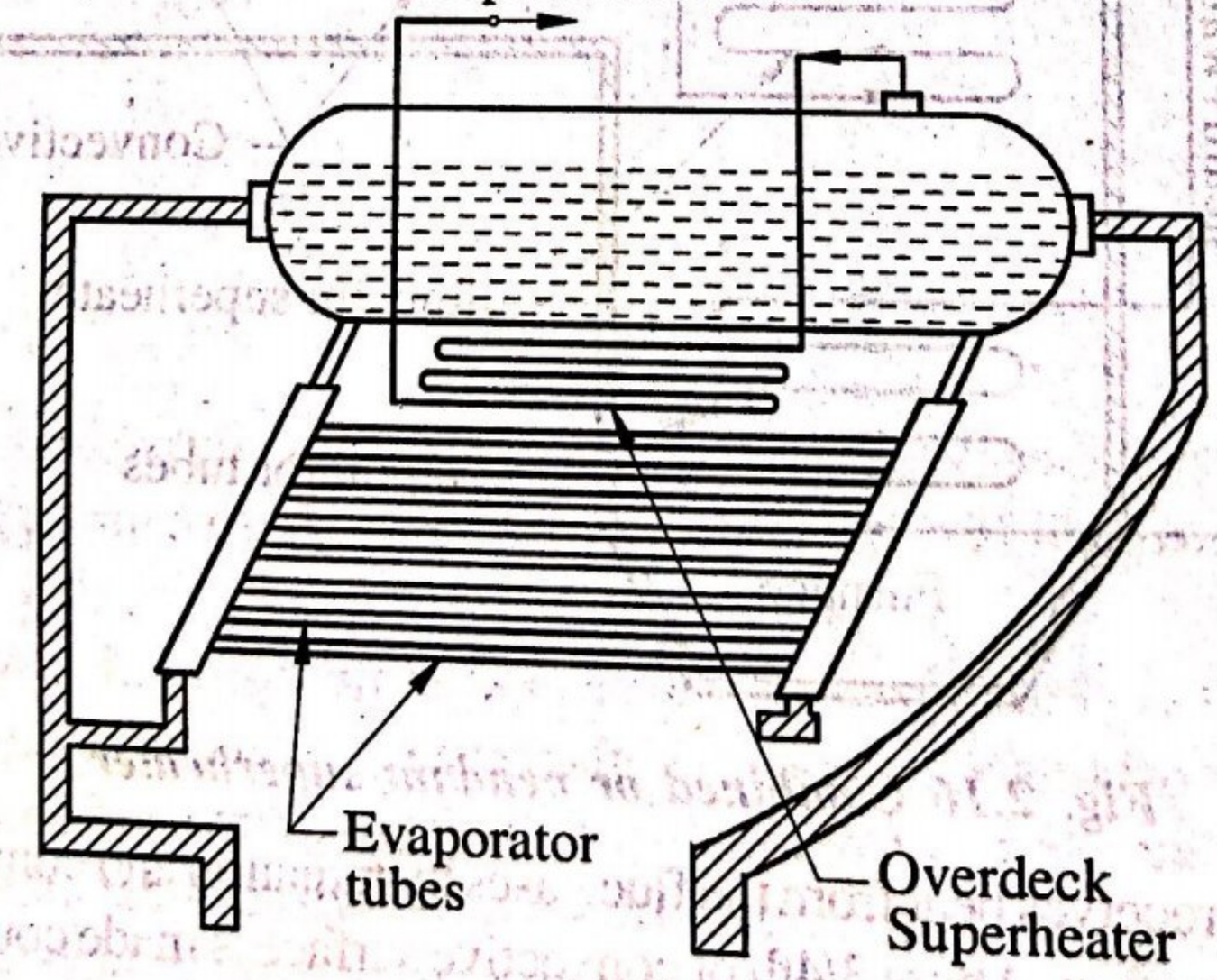


Fig. 2.17(b) Overdeck superheater

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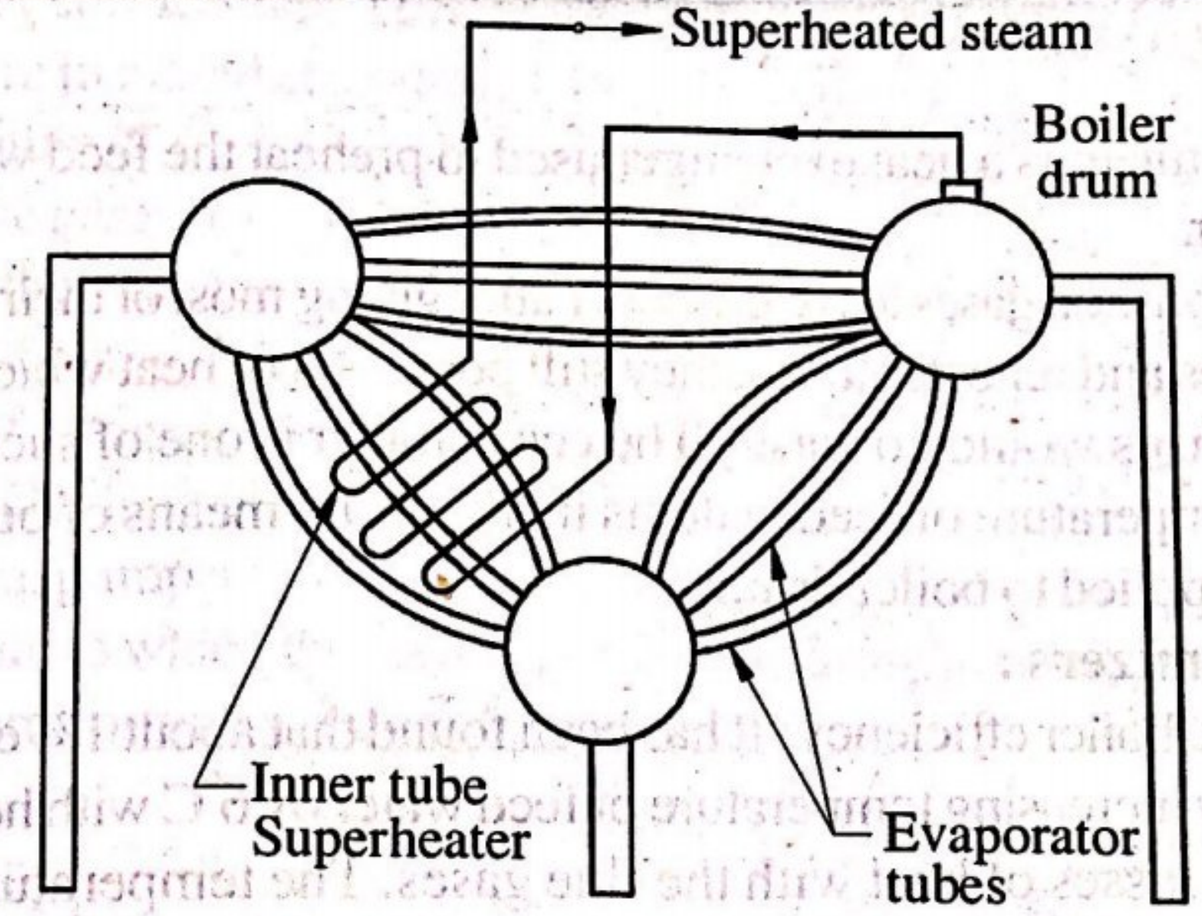


Fig. 2.18(a) Inner tube superheater

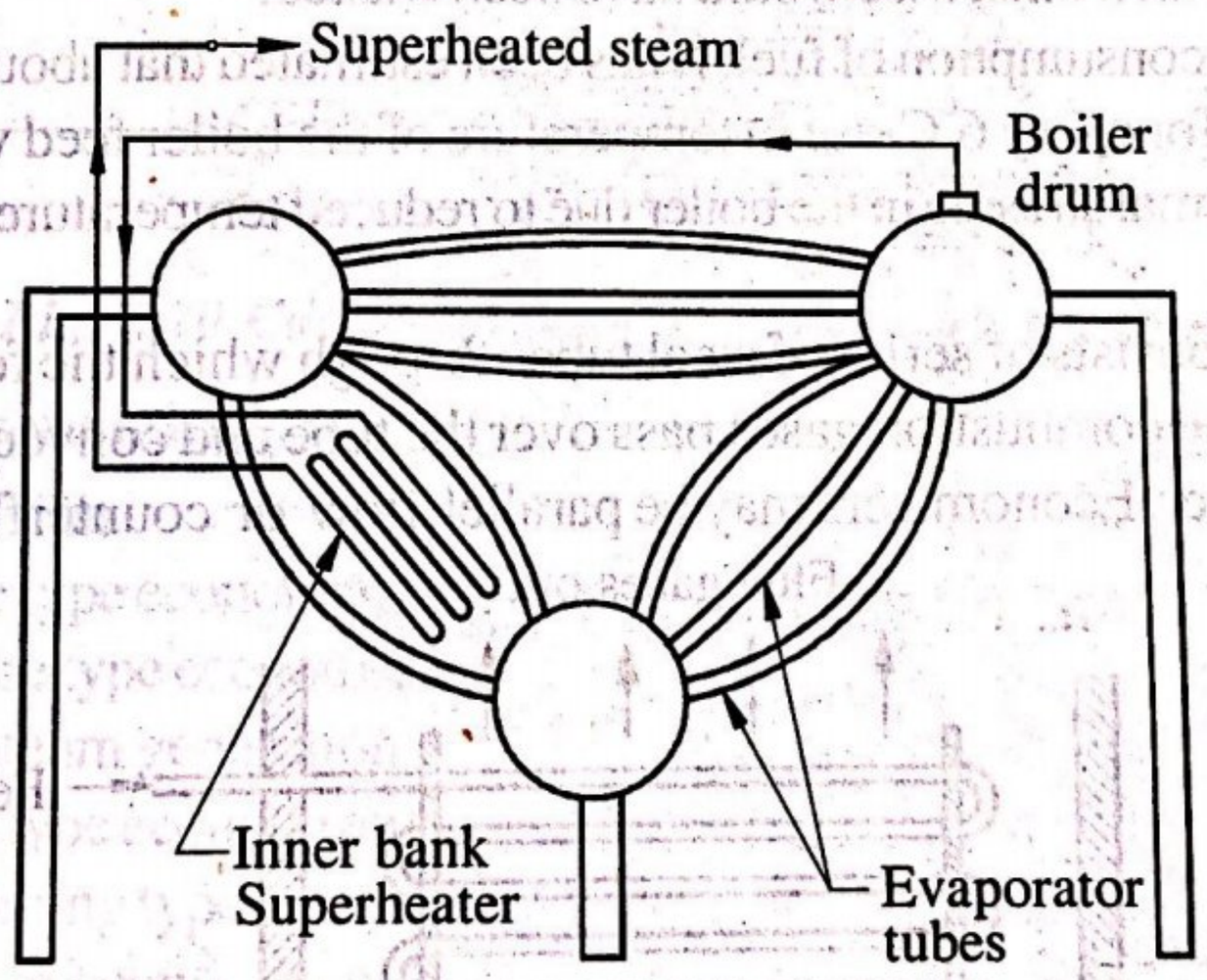


Fig. 2.18(b) Inner bank superheater

2.12 Reheaters

Function : The function of the re-heaters is to resuperheat the partly expanded steam from the turbine. This is done so that the steam remains dry as far as possible through the last stage of the turbine.

The reheater is generally located before or after the convective superheater in the convective zone of utility boilers. In modern high pressure boilers, reheaters are generally in two sections. The primary section is placed in the convective zone and secondary section is placed just at the furnace exit hanging from the top. The design consideration for reheaters are similar to those for superheaters. It having same output temperatures but steam pressures are about 20-25% of those in superheaters. Hence lower grade steel alloy can be used for reheaters.

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2.13 Economizers

Function : The economizer is a heat exchanger used to preheat the feed water using the hot gases leaving the boiler.

When the combustion gases leave the boiler after giving most of their heat to evaporator tubes, superheater tubes and reheater tubes, they still posses lot of heat which if not recovered by means of some devices would go waste. The economizer is one of such heat recovering device in which the temperature of feed water is increases by means of outgoing flue gases before feed water is supplied to boiler drums.

Advantages of economizers :

(1) It improves the boiler efficiency. It has been found that about 1% efficiency of boiler is increased by increasing temperature of feed water by 6°C with help of economiser.

(2) It reduces the losses of heat with the flue gases. The temperature of flue gases is about 370°C to 540°C at exit of last superheater or reheater, having large amount of heat energy which otherwise would have been wasted.

(3) It reduces the consumption of fuel. It has been estimated that about 1% of fuel costs can be saved for every 6°C rise in temperature of the boiler feed water.

(4) It reduces thermal stresses in the boiler due to reduced temperature differential in the boiler.

Economizer consists of series of steel tubes through which the feed water flow as shown in Fig. 2.19. The combustion gases pass over the tube and convective heat transfer taken place to feed water. Economizers may be parallel flow or counterflow, when the gas

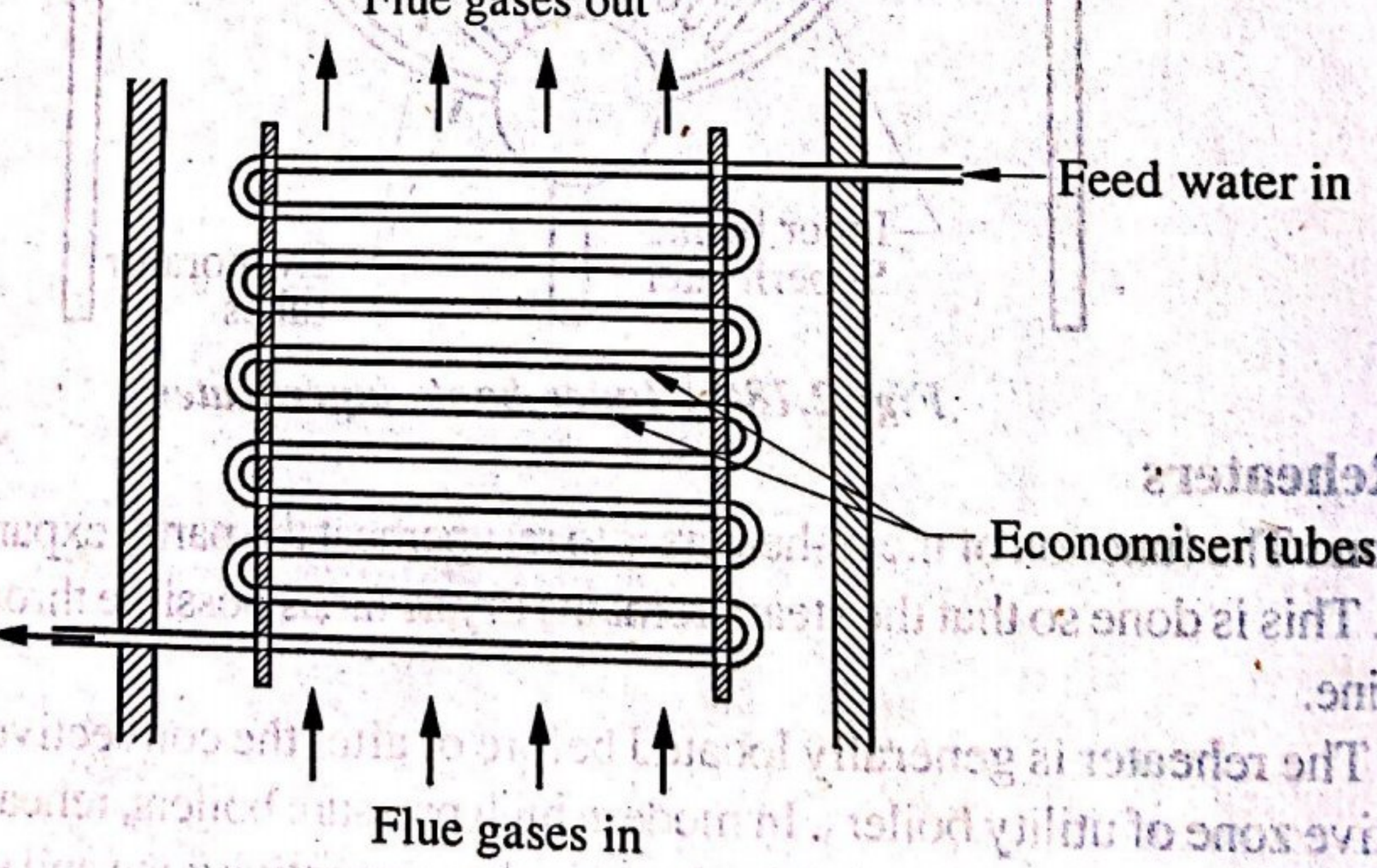


Fig. 2.19 Economizer

flow and water flow are in same direction the economizer are called parallel flow whereas in counter flow the gas flow and water flow in the opposite direction. A counter flow arrangement is normally selected so that heating surface requirement is kept minimum for the same temperature drop in flue gas. Economizer tubes are made of steel either smooth or covered with fins to

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increase heat transfer surface area. Economizer tubes are generally 45 to 70 mm in outside diameter and coils are installed at a pitch of 45 to 50 mm spacings.

The temperature of feed water entering the economizer should be high enough so that moisture from the flue gases does not condense on the economizer tubes which may be absorb SO_2 and CO_2 from the flue gases and form acid to corrode the tubes. Also the feed water temperature at inlet of pump should be less than 85°C. Since there is a slight negative pressure on the suction side of pump, if temperature of feed water is greater than 85°C steam bubbles begin to form and boiler feed pump will bot be able to pump mixture of steam and water and flow stops. Therefore, pump always place before the economizer as shown in Fig. 2.20. The maximum temperature to which the feed water can be heated in the economizer is 25°C below saturation temperature corresponding the pressure.

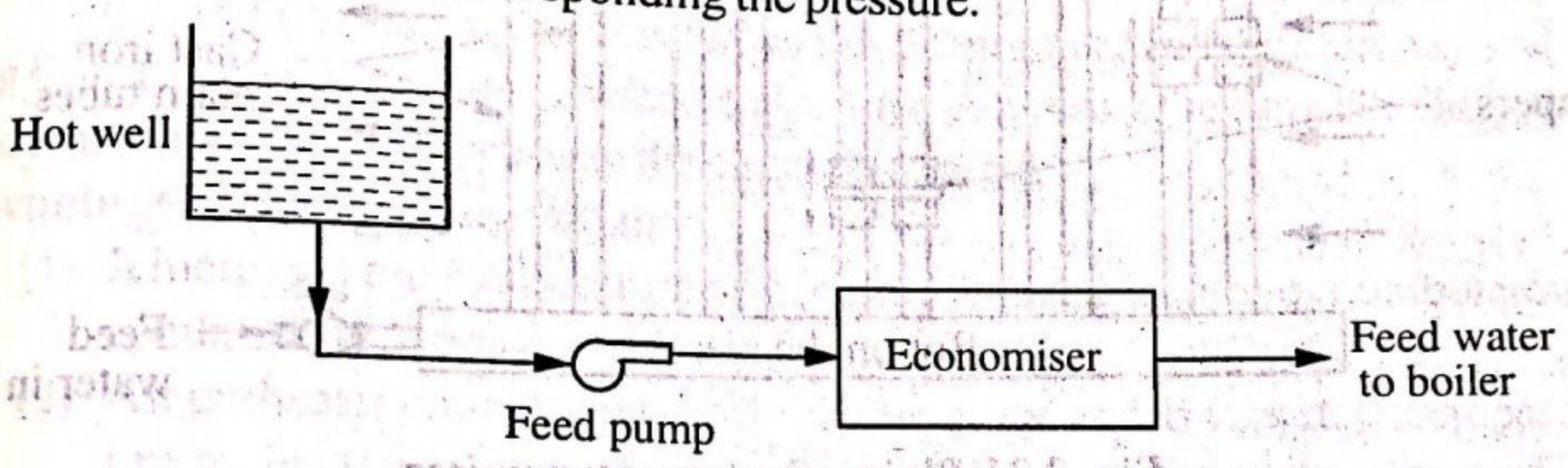


Fig. 2.20 Correct position of economizer & pump

Classification of economizers :

1. Based on construction
   - (i) Plain tube type economizers
   - (ii) Gilled tube type economizers
2. Based on part of steam generation
   - (i) Steaming type economizers
   - (ii) Non-steaming type economizers
3. Based on location of economizers
   - (i) Independent economizers
   - (ii) Integral economizers

Plain tube type economizer : In this economizer, the group of tubes are made of cast iron to resist corrosive action of flue gases. The ends of tubes are pressed into top and bottom headers. The feed water is distributed along the section by branched pipe coupled to the bottom headers. The economizer is located in the main flue between the boiler and the chimney. The waste flue gases flow outside the economizer tubes and heat is transferred to the feed water flowing inside the tubes.

The external surface of tubes is kept clean and free from soot by soot scrapers moving up and down the economizer tubes. Otherwise heat transfer resistance is increase and efficiency of economizer decreased. Every economizer is provided with a safety valve, a drain valve and

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air release valve. These types of economizer are generally used in Lancashire boiler working under natural draught.

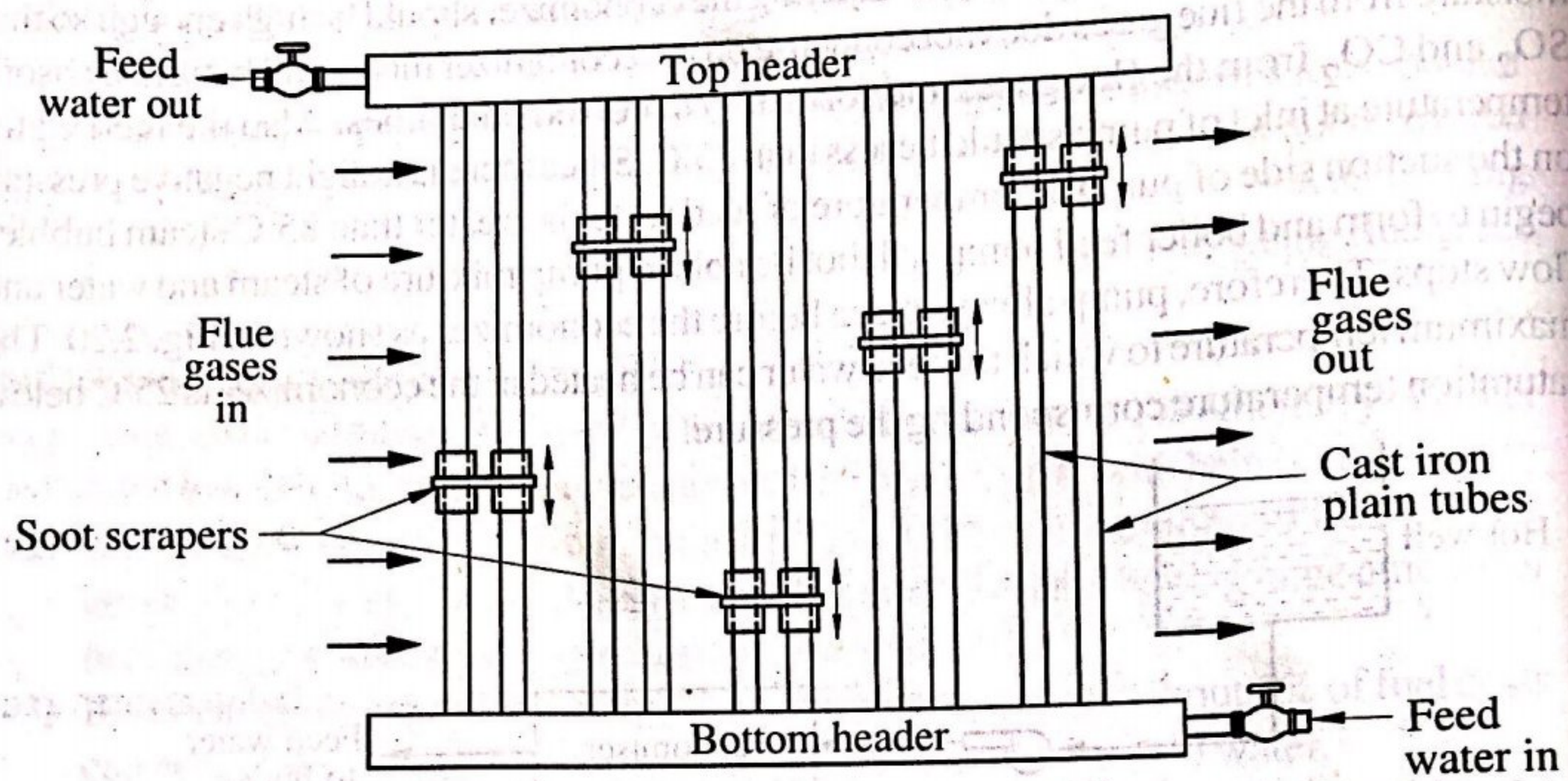


Fig. 2.21 Plain tube type economizer

Gilled tube type economizer : In this economizer, casting rectangular gills are provided on the bare tube walls. Hence, heat transfer surfaces are provided on the bare tube walls. Therefore, heat transfer surface are increased and size of economizer is reduced. Cast iron gilled tube economizers can be used upto 50 bar working pressure. At higher pressures (> 50 bar) steel tubes are used instead of cast iron but cast iron gilled sleeves are shrunk to them.

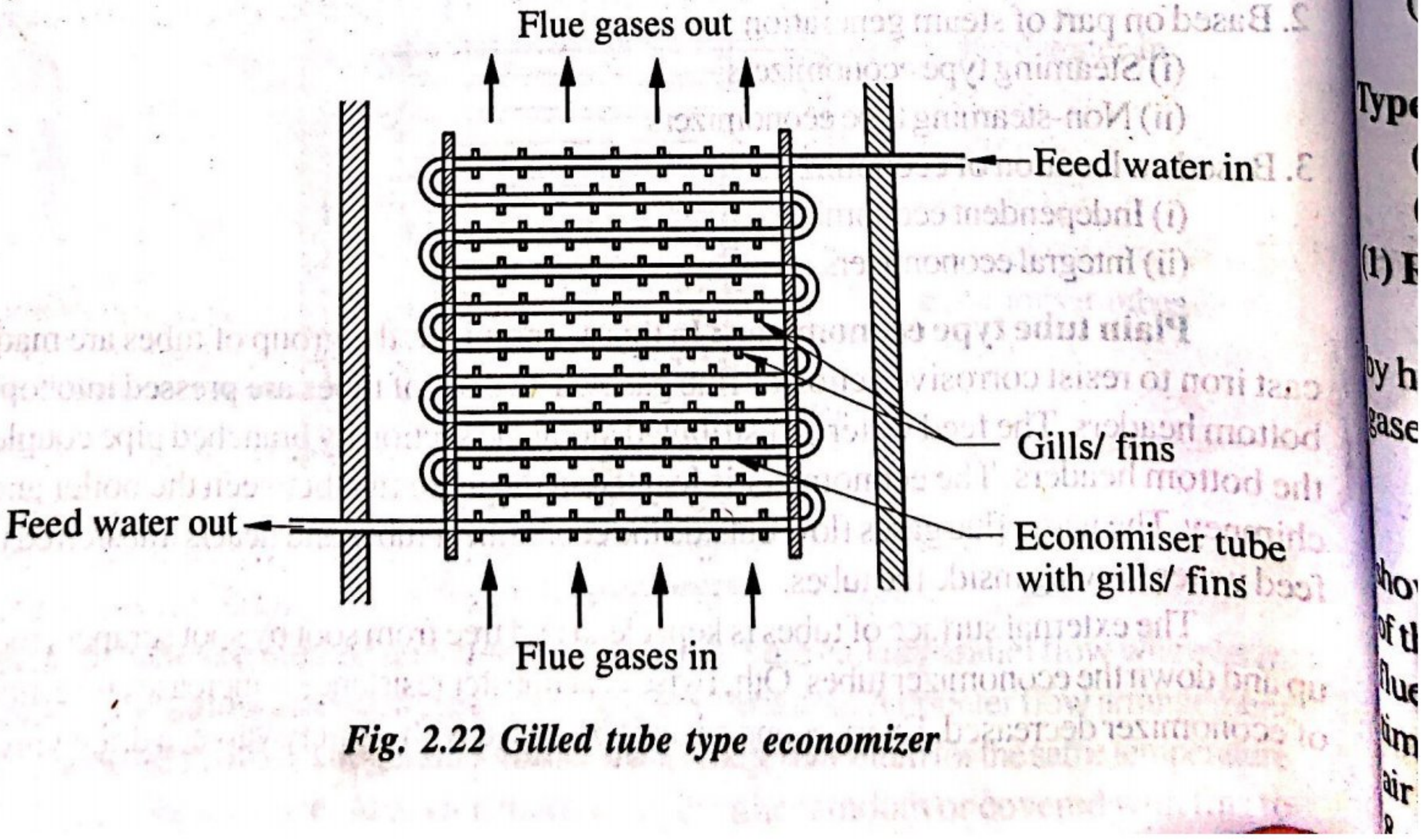


Fig. 2.22 Gilled tube type economizer

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In case of **steaming economizer**, some part of the water (about 5 to 7%) to be converted into steam during its passage through the economizer. This types of economizer are used in large steam power plants. While in case of non steaming economizer feed water is heated within 75% of the saturation temperature of the boiler.

When economizer is installed outside the boiler house at any convenient place then it is known as *independent economizer*. While *integral economizer* is installed close to the boiler.

2.14 Air pre-heaters

Function : The air preheaters are employed to recover the heat from the flue gases and this heat is utillized to increased the temperature of air before it supply to the furnace.

An air preheater is placed between the economizer and the chimney and it extracts heat from the flue gases and transfers to air which is entering the furnace. The portion of the heat that otherwise would pass up the chimney to waste.

Advantage of pre-heating the air :

(1) It increases the temperature of the furnace gases, improves combustion rates and combustion efficiency.

(2) Air preheater extracts heat from the flue gases, and lowers stack temperatures thus improving the overall efficiency of the boiler. It has found that a drop of 20-22°C in the flue gases temperature increases boiler efficiency by 1%.

(3) The preheating air facilitates the burning of poor grades of fuel thus permitting a reduction in excess air.

(4) It increases steam generating capacity per unit m^2 of boiler surface.

(5) Increased thermal efficiency of plant and saving of fuel.

(6) In case of pulverised coal combustion hot air can be used to dry coal, to transport pulverized coal to burners and to accelerate combustion.

Types of air preheaters :

(1) Recuperative type : (i) Plate type (ii) Tubular type

(2) Regenerative type

(1) Recuperative type air preheaters

In recuperative type of air preheaters, the two fluids (air and flue gases) are separated by heat transfer surface, one fluid (air) flowing constantly on one side and other fluid (flue gases) on other side of the surface.

Recuperative type of air preheater is further classified as tubular type and plate type.

Tubular air preheater : It consists of a large number of tubes rolled into sheets as shown in Fig. 2.23. The flue gases flow through tubes and air is passed over the outer surface of the tubes in a direction opposite to that of flue gases flow. The heat transfer takes place from flue gases to air to be supplied to the furnace. The horizontal baffles are provided to increase time of contact which will help for higher heat transfer. A soot hopper is fitted to the bottom of air heater casing to collect soot. The diameter and height of tubes are commonly used for 6 to 8 cm and 3 to 10 m respectively.

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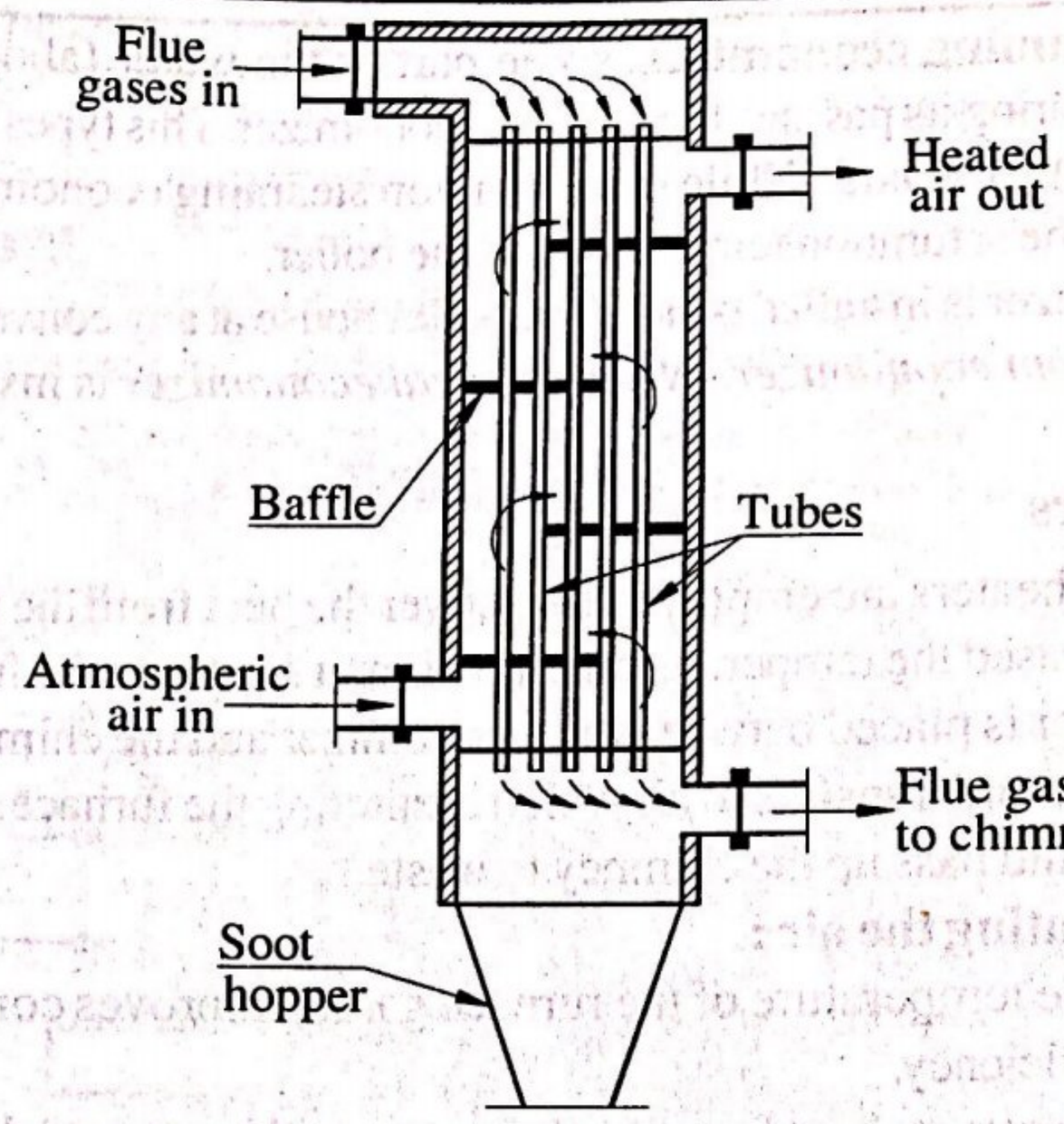


Fig. 2.23 Tubular air preheater

Plate type air preheater : It consists of rectangular flat plates spaced from 1.25 cm to 2.5 cm apart formed alternate flue gases and air passages as shown in Fig. 2.24. This type of air preheater is more expensive for installation and maintenance compared to tubular type, hence it is not used in modern power plants.

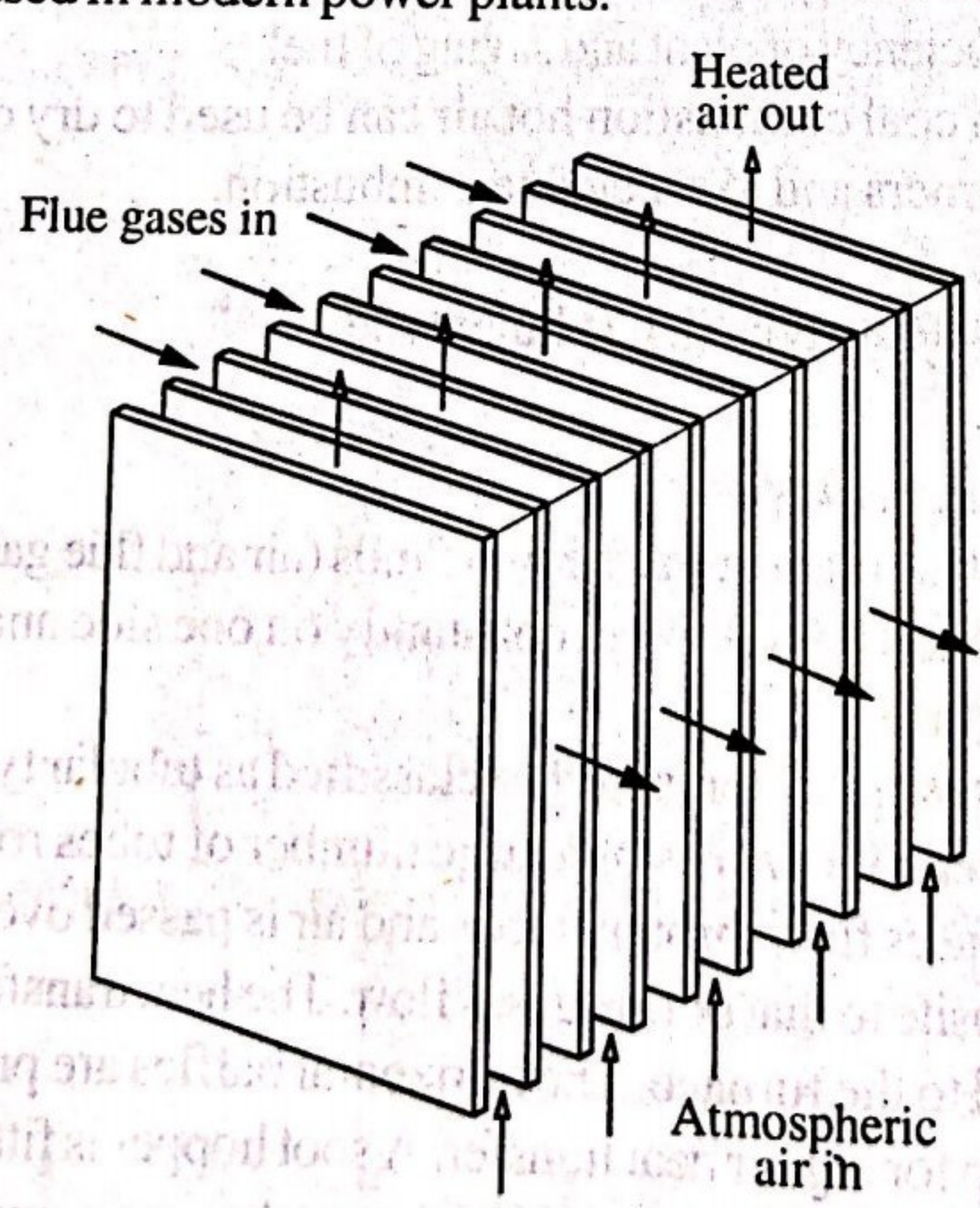


Fig. 2.24 Plate type air preheater

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(2) Regenerative air preheaters

Regenerative type air preheater has a energy storage medium called the matrix which is alternately exposed to the hot and cold fluids. When the hot flue gases flow through the matrix in the first half of the cycle the matrix gets heated and the flue gases cooled. In the next half of the cycle when air flows through the matrix air gets heated and matrix is cooled. The cycle is repeated. The heat transfer from flue gases to air is thus periodic. There are two types of regenerative air preheaters as stationary regenerative air preheater and rotary regenerative air preheater.

A rotary regenerative air preheater is shown in Fig. 2.25 called ***Ljungstrom air preheater,*** which consists of a rotor driven by an electric motor. The rotor divided into a number of sectors (12 to 24 sectors), each sector being fixed with steel sheets, either flat or corrugated. These sheets form the heat transfer surfaces. The rotor is placed in a drum which has been divided into two compartments, air and flue gases compartments. To avoid leakage from one compartment to the other seals are provided. The rotor rotates at a very slow speed of 3-4 rpm. As the rotor rotates, it alternately passes through flue gases and air zones. The rotor sheets are heated by the flue gases in their zone and transfer this heat to air when they are in air zone.

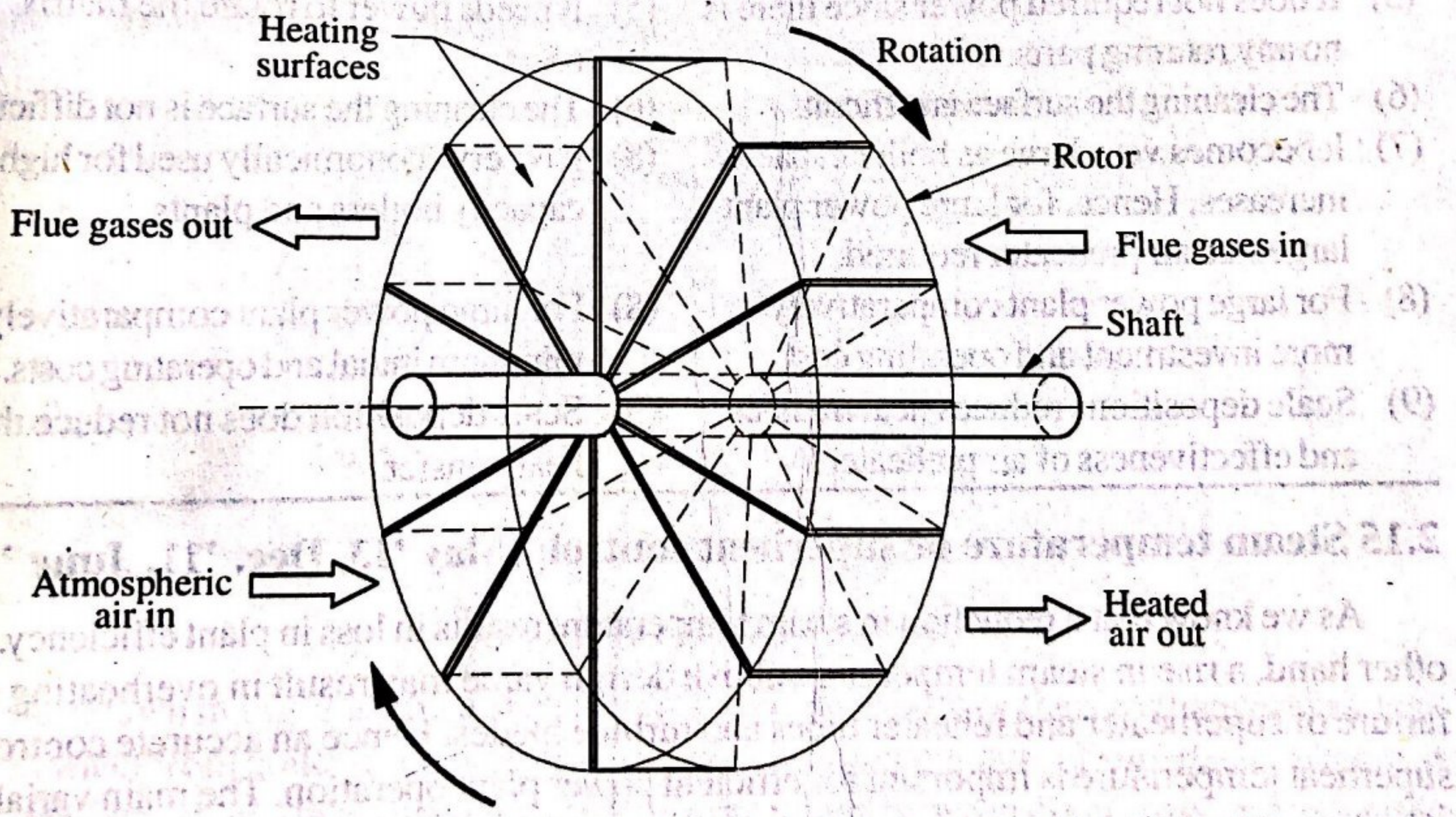


Fig. 2.25 Regenerative type (Ljungstrom) air preheater

The regenerative type air preheater are compact, have less weight, occupy less space and provide enormous amount of heat transfer surface and are therefore widely used in steam power plants.

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Comparison between recuperator and regenerative air pre-heaters :

Recuperator air preheater	Regenerative air preheater
(1) In this heat exchanger two fluids are separated by heat transfer surface. There is continuously heat transfer takes place.	(1) In this heat exchanger cold fluid and hot fluid alternately occupy same surface. There is discontinuously heat transfer takes place. First half time of cycle surface heated by gas and remain half cycle same surface cooled by air.
(2) Static construction, there is only nominal leakage through joints.	(2) Heat transfer surfaces are rotated, hence leakage through sealing arrangement at the moving surfaces is high.
(3) It requires more space compared to regenerative type air preheater.	(3) Large heat transfer surfaces being accommodated in a small volume. Hence it is very compact air preheater.
(4) Less pressure drops for both air and flue gases.	(4) Large pressure drops for both air and flue gases.
(5) It does not required power since there is no any rotating parts.	(5) It needs power to rotate the matrix rotor.
(6) The cleaning the surface is difficult.	(6) The cleaning the surface is not difficult.
(7) It becomes very large as boiler capacity increases. Hence, for large power plant large size air preheater required.	(8) It is very economically used for high capacity boilers and plants.
(8) For large power plant comparatively more investment and operating cost.	(8) For large power plant comparatively minimum initial and operating costs.
(9) Scale depositions reduces heat transfer and effectiveness of air preheaters	(9) Scale deposition does not reduce the heat transfer.

2.15 Steam temperature or superheat control [May '13, Dec. '11, June '11]

As we know that a reduction in steam temperature results in loss in plant efficiency. On other hand, a rise in steam temperature above design value may result in overheating and failure of superheater and reheater tubes and turbine blades. Hence an accurate control of superheat temperature is important for efficient power plant operation. The main variables affecting superheat temperature are as furnace temperature, mass flow rate of flue gases, cleanliness of heating surfaces (superheaters) and variation of load on the plant. Out of these effect of load variation on superheat temperature is most important. The superheat of steam can be controlled by following methods.

(1) Combined radiant - convective superheaters : As we know load increases, the heat transfer in the radiant superheater decreases and heat transfer in convective superheater increased as shown in Fig. 2.16. To keep the temperature constant, radiant and convective superheater

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are arranged in series as shown in Fig. 2.26.

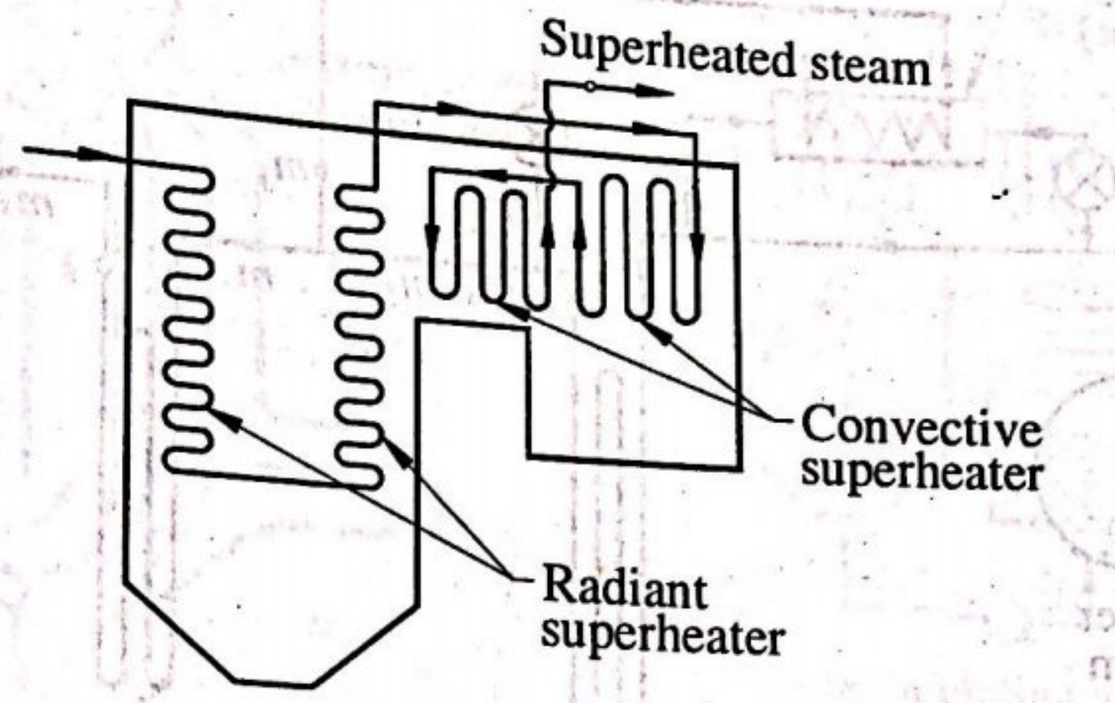


Fig. 2.26 Radiant and convective superheater in series

(2) Desuperheating and attemperation [June '11] : The temperature of steam is reduced by removing energy from the steam and is known as attemperation. A spray type attemperator as shown in Fig. 2.27(a) is called desuperheater in which low temperature water (from feed water) is injected into pipe carrying steam. This causes to vaporise water, absorbs heat from the steam and reduces temperature of steam.

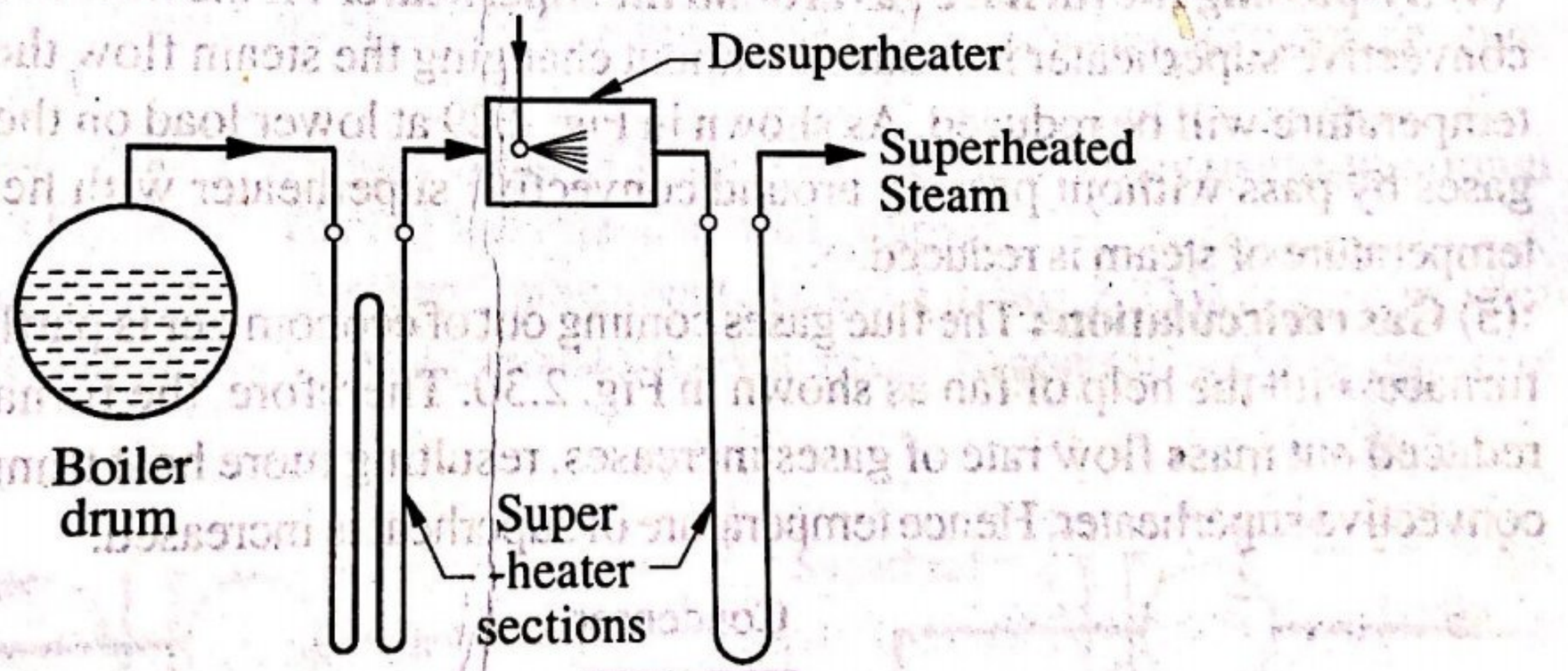


Fig. 2.27(a) Desuperheating

As shown in Fig. 2.27(b), a shell and tube type attemperator (heat exchanger) in which boiled water from boiler drum is utilized to reduced the temperature of superheated steam. A portion of steam (m_1) taken out through tubes from a point between the primary and secondary superheaters by an automatic valve diverted to heat exchanger where boiled water from boiler drum is circulated. The boiled water coming from drum absorbs heat from superheated steam and finally, boiled water remixes with the primary steam before entering the secondary superheater.

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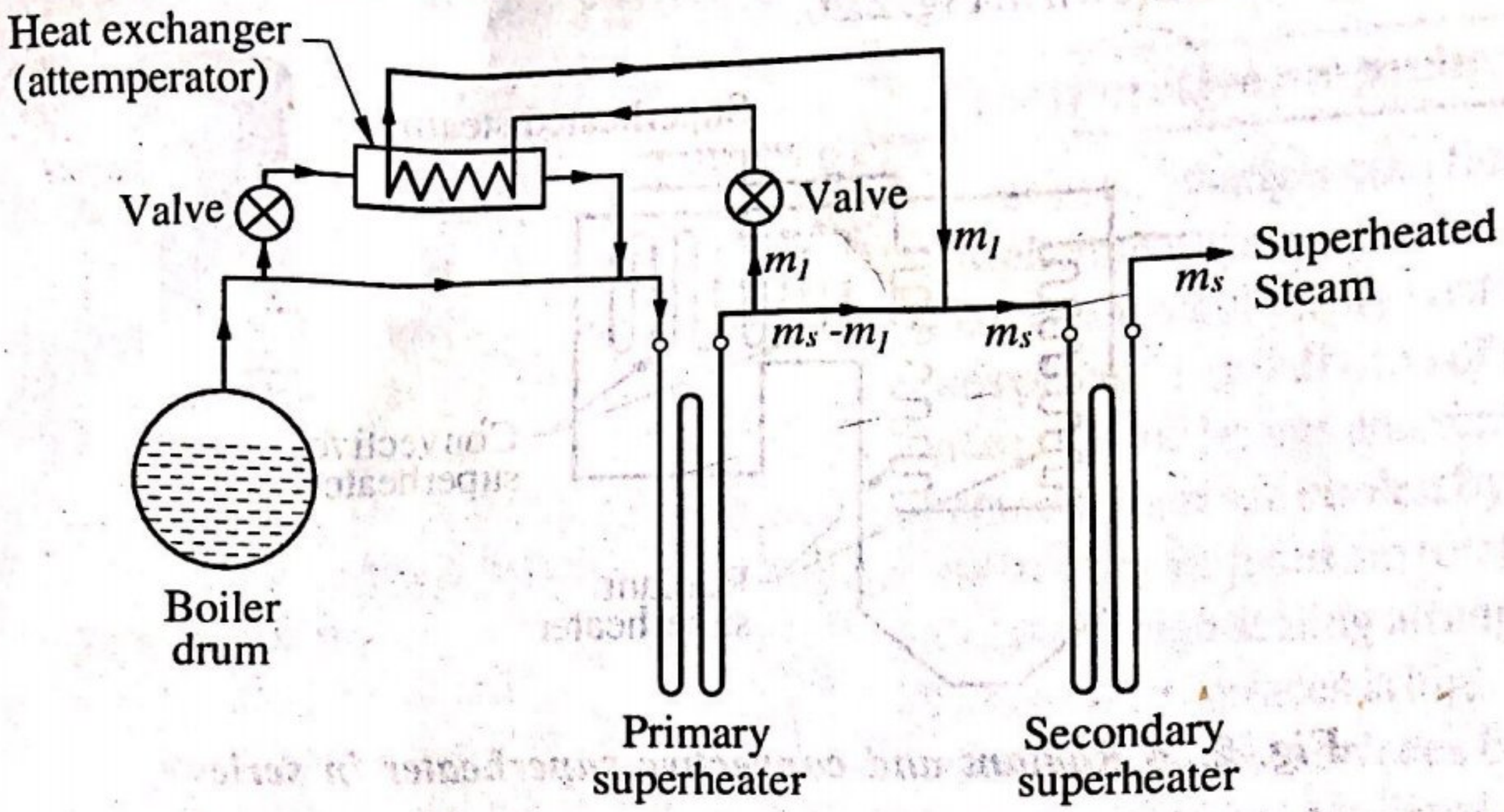


Fig. 2.27(b) Attemperation

(3) Pre-condensing the steam : The temperature of the steam can be controlled by condensing the steam coming out of boiler with a small condenser with the help of feed water as shown in Fig. 2.28. Automatic control regulates the amount of feed water by-passed.

(4) By-passing the furnace gas around the superheater : If the flow of flue gases through convective superheater is reduced without changing the steam flow then the final steam temperature will be reduced. As shown in Fig. 2.29 at lower load on the plant, the part of gases by pass without passing around convective superheater with help of damper, the temperature of steam is reduced.

(5) Gas recirculation : The flue gases coming out of economizer is partly recirculated into furnace with the help of fan as shown in Fig. 2.30. Therefore, the furnace temperature is reduced but mass flow rate of gases increases, resulting more heat transfer takes place in convective superheater. Hence temperature of superheat is increased.

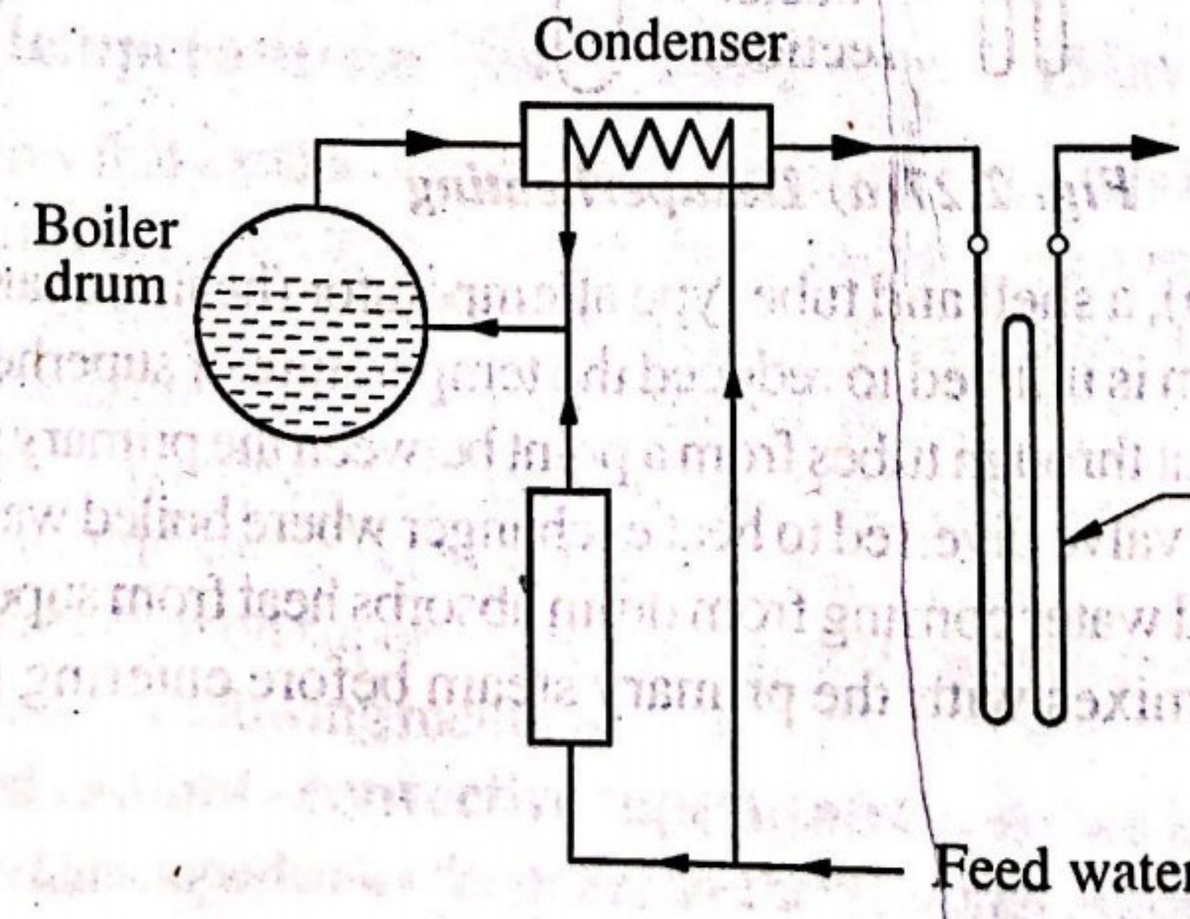
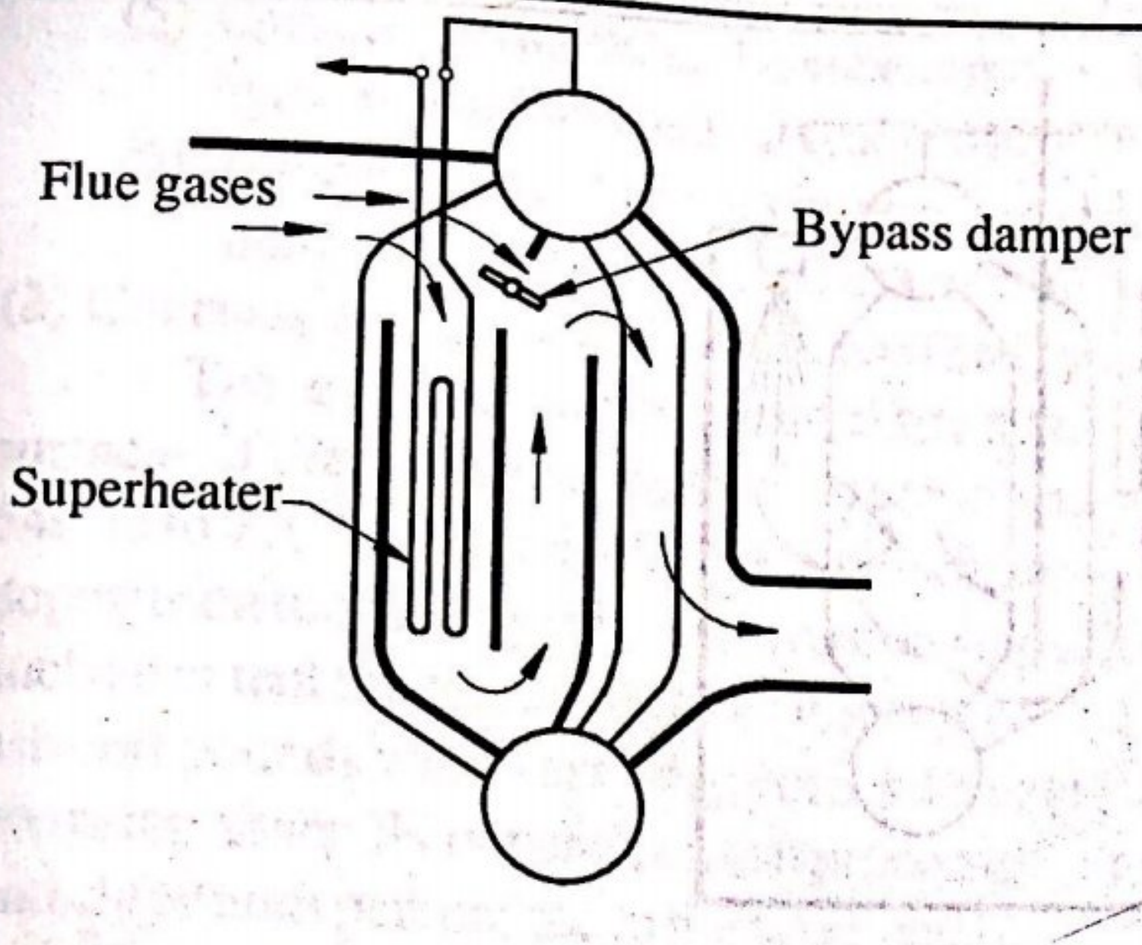


Fig. 2.28 Pre-condensing the steam

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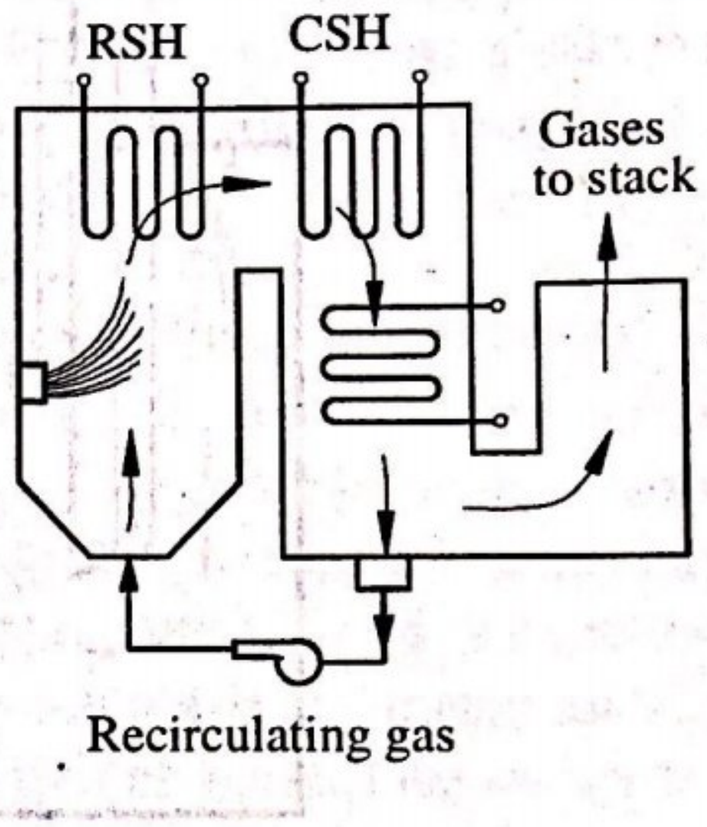


Fig. 2.29 By pass control *Fig. 2.30 Gas recirculation*

(6) Tilting burners in the furnace : The temperature of the steam can be control by changing heat absorption through superheaters and evaporator tubes. As shown in Fig. 2.31 by tilting the burner downward in a furnace, more heat of the gases available around the evaporator tube and less heat available at superheater, hence temperature of superheated steam is reduced. If the burner is turned upward more heat of flue gases available at superheater, hence temperature of steam is increased.

(7) Auxiliary burners : The temperature of steam can be controlled by using additional burner as shown in Fig. 2.32, effect of this similar to tilting burner.

(8) Twin furnace : The twin furnace arrangement as shown in Fig. 2.33 is an extension of the separately fired superheater. The superheat temperature can be control by changing rate of firing.

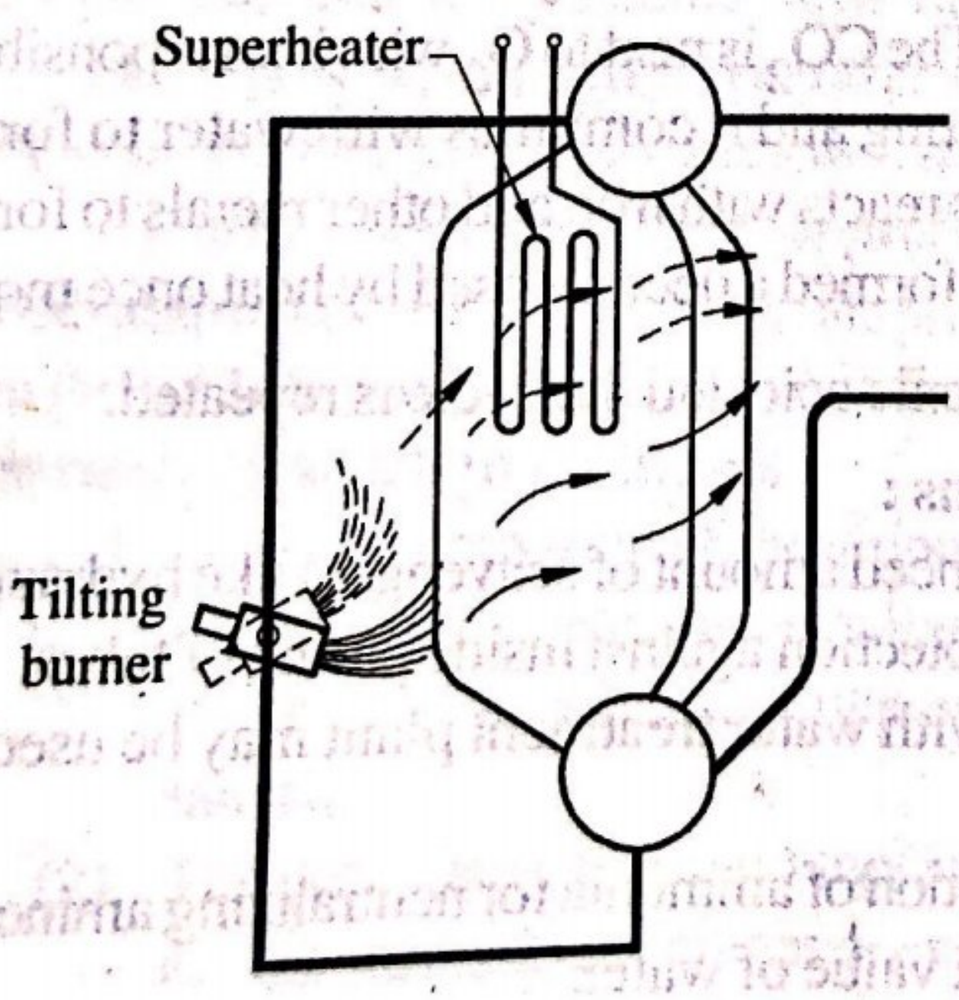


Fig. 2.31 Tilting burner.

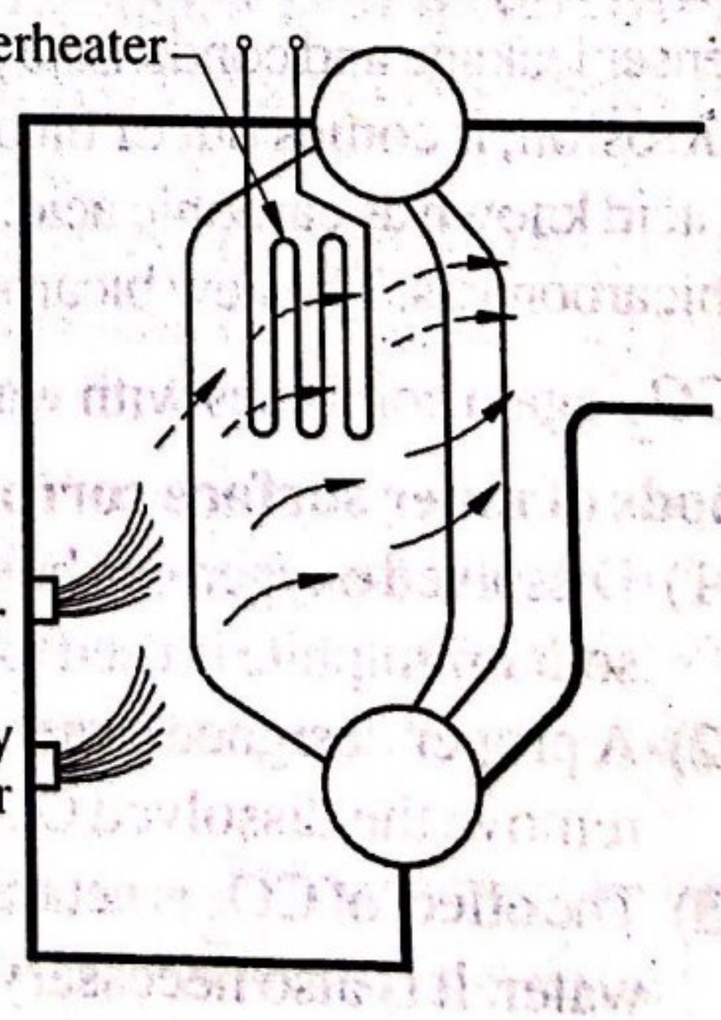


Fig. 2.32 Auxiliary burner

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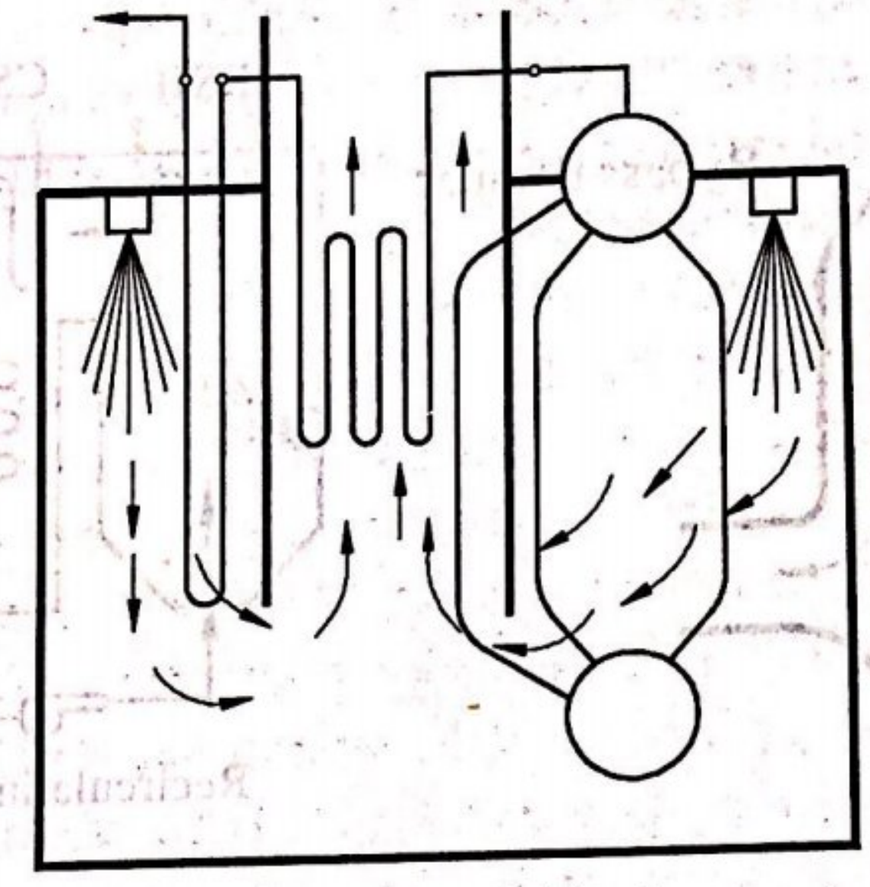


Fig. 2.33 Twin furnace

2.16 Corrosion in boilers and its prevention [Dec. '13, Dec. '11]

Corrosion is the destructive conversion of metal into oxides or salts. Simply, corrosion means the loss of materials. Corrosion may occur in the boiler shell, evaporating tubes, superheater, economiser, air preheaters. Corrosion occurs in inner surface of boiler tubes due to acidity present in water. Corrosion takes place on external surface (gas side) of tubes due to coal ash. Corrosion in power plant equipment produces pits, grooves and cracks or a general wastage of the wall material. Allowed to continue, corrosion ultimately makes metal parts fail.

(1) Corrosion at inner surfaces (water/steam side)

The corrosion is caused by (i) an acid or low pH value in water and (ii) the presence of O_2, CO_2 or chlorides dissolved in feed water. The presence of O_2 is mostly responsible for corrosion among other factors. Oxygen generally enters a closed system through make up condenser leakage and condensate pump packings. The CO_2 is next to O_2 which is responsible for corrosion, it comes out of bicarbonates on heating and it combines with water to form weak acid known as carbonic acid. This acid slowly reacts with iron and other metals to form their bicarbonates. The new bicarbonates of metals formed a decomposed by heat once more and CO_2 again combines with water to form carbonic acid and the cycle is repeated.

Methods of inner surface corrosion preventions :

(1) Dissolved oxygen can be removed by balanced amount of scavengers like hydrazine sodium sulphite is used to provide total protection against inside pitting of tubes.

(2) A proper designed deaerator combined with water treatment plant may be used to remove the dissolved O_2 and CO_2.

(3) The effect of CO_2 is neutralized by the addition of ammonia for neutralizing amines in water. It is also necessary to lower the pH value of water.

(4) The effect of acid is neutralized by the addition of alkali salts in water.

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(5) Carbon dioxide (CO_2) can be removed by adding balanced quantity of ammonium hydroxide in the water. It reacts with CO_2 and form ammonium carbonate and water.

(6) The corrosion of metal surfaces can be prevented by applying protective coating of amines to the internal surfaces of boilers tubes.

(2) Corrosion at external surfaces (flue gases side)

The external corrosion coal fired boiler is mainly caused by coal ash. The outer surfaces of the tube are subjected to ash corrosion when the temperature range is between 540°C to 710°C. Alkali sulphates which are formed at high temperature in vapour form, deposits on tube surfaces and corrosion starts. This type of corrosion reduces tube walls to such an extent that metal ruptures under the working pressure. Certain coals produces liquid ash compounds which are very corrosive to all conventional boiler materials. The external corrosion generally depends on ash properties, rate of ash deposition, tube surface temperature and chromium percentage in the tube material.

External corrosion in the ***superheaters*** and ***reheaters*** of coal fired unit is due to coal ash. In coal ash corrosion the low melting species would be sodium or potassium iron trisulphates [$Na_3Fe(SO_4)_3$ or $K_3Fe(SO_4)_3$]. High temperature (about 550°C) of superheater and reheaters favour the formation of these low-melting compounds. The corrosion rate in coal ash corrosion between 600°C to 680°C is very high. In this temperature range Na_2SO_4 or K_2SO_4 reacts with surface oxides in the presence of SO_3 to form complex liquid sulphates [$Na_3Fe(SO_4)_3$ or $K_3Fe(SO_4)_3$] that result in rapid corrosion. These corrosion is known as high temperature corrosion.

External corrosion of the ***economizer*** tubes is very serious when high sulphur content wet fuels are used in the boiler furnace as the chances of forming H_2SO_4 are more. External corrosion also occurs if flue gases contains water vapour, which condenses at the tube surfaces. Sulphurous acid is formed when SO_2 is dissolved in free moisture in the flue gases, sulphur trioxide may combine with superheated water vapour to form H_2SO_4 at the acid dew point. This reaction normally occurs between 115°C to 140°C. Hence minimum feed water temperature passing through economizer should be higher than this range of dew point of acid. Hence this corrosion is known as low temperature corrosion.

The external corrosion of ***air preheater*** is similar to that of economizer. However, the temperature of air passing through air preheater is less than the acid dew point temperature then the acid corrosion is more serious than economizer.

External corrosion preventions :

(1) To avoid corrosion of the air preheater, the flue gases should not be cooled below the dew point of corrosion species. This can be achieved by passing some air around the preheaters or by recirculating air from the preheater outlet to the forced draught fan inlet.

(2) The deposits of scale and soot should be regularly removed from external surfaces of economizer, superheater, evaporator tubes, air preheaters, reheaters etc.

(4) If the low inlet temperature of the feed water is the cause of low temperature corrosion,

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it can be corrected by heating the water with the help of steam from the boiler in the shell and tube heat exchanger.

(5) If low sulphur coal is used where chances of carrying sulphur trioxide with the gases are less then there is possibility to use low temperature of feed water at inlet with an adequate margin is held above the acid dew point.

(6) The high temperature corrosion problem can be solved by using coal having more favorable characteristics, providing stainless steel tube shields, and replacing the damaged tubes with tubes containing high chromium content.

Problem 2.1 *The following data refer to a boiler plant consisting of an economiser, a boiler and a superheater.*

Mass of water evaporated per hour = 5600 kg
Mass of coal burnt per hour = 650 kg,
Calorific value of coal = 31500 kJ /kg,
Steam pressure = 14 bar,
Dryness fraction of steam leaving the boiler = 0.95
Temperature of steam leaving the superheated - 250° C.
Temperature of feed water entering the economiser = 30° C
Temperature of feed water leaving the econimiser = 120° C
Specific heat of water = 4.1968 kJ/kg K
Specific heat of steam = 2.3 kJ/kg K

Determine : (i) Percentage of heat of fuel utilized in economiser, boiler and superheater. (ii) Overall efficiency of plant. (iii) Equivalent evaporation (cinsider super heater is part of boiler) ***[Same as Jan. 2013]***

Solution

Given data : $m_s = 5600\ kg/hr$, $m_f = 650\ kg/hr$,

C.V. = 31500 *kJ./kg*, $p = 14$ bar, $x = 0.95$

$t_{e_1} = 30^\circ C$, $t_{e_2} = 120^\circ C$, $T_{sup} = 250^\circ C$

$c_{pw} = 4.1868\ kJ/kg\ K$, $c_{ps} = 2.3\ kJ/kg\ K$,

heat utilized = ?, η = ?, m_{SE} = ?

From steam table, at pressure p = 14 bar

$h_f = 830.07\ kJ/kg$, $h_g = 2787.8\ kJ/kg$, $T_s = 195.04\ ^\circ C$

Enthalpy of steam leaving the boiler,

$$h_2 = h_f + x(h_g - h_f)$$
$$= 830.07 + 0.95(2787.8 - 830.07)$$
$$= 2689.91\ kJ/kg$$

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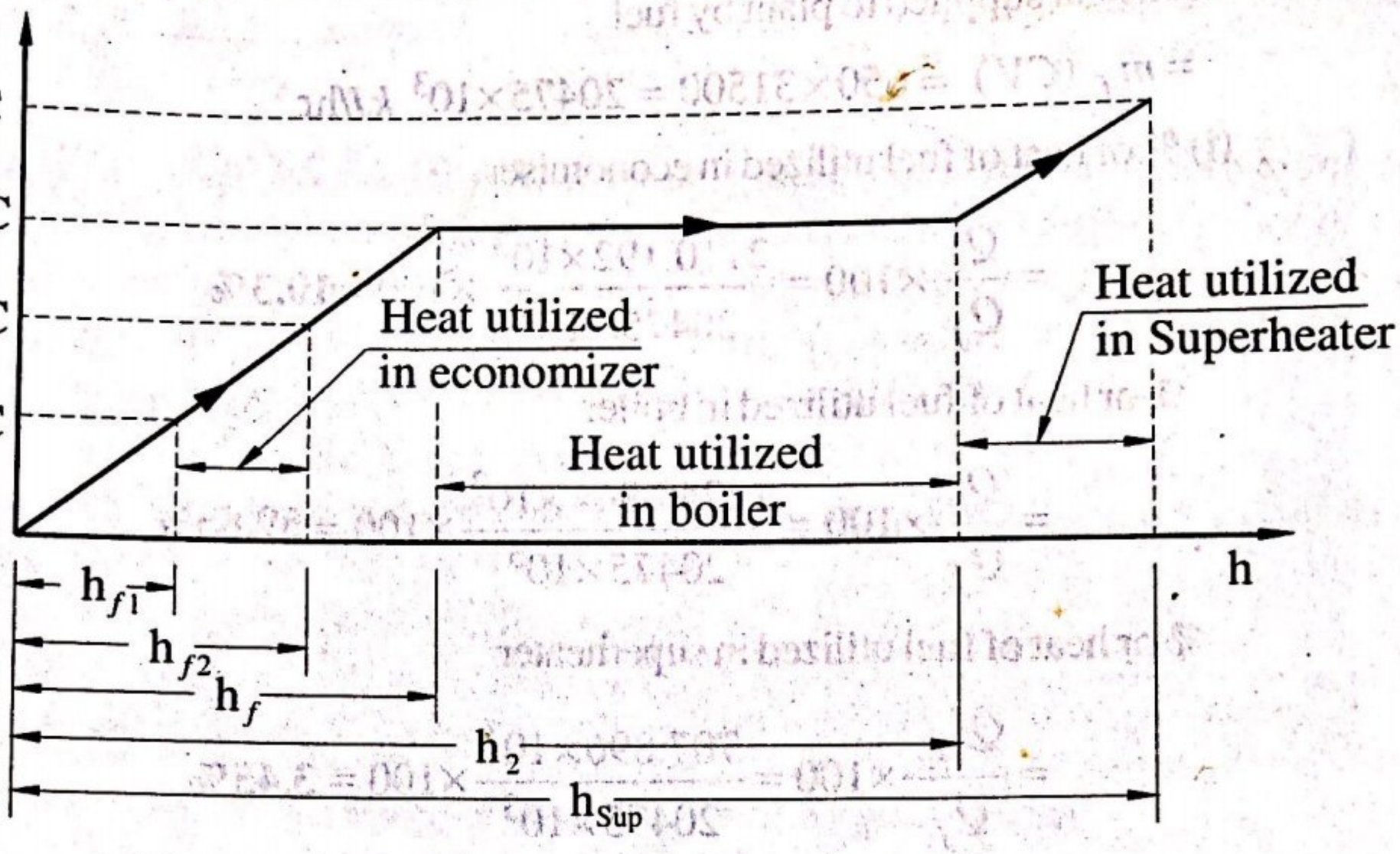


Fig. 2.34 T-h diagram for steam formation

Enthalpy of steam leaving superheater

$$h_{sup} = h_2 + C_{ps}(T_{sup} - T_s)$$
$$= 2689.91 + 2.3\ (250 - 195.04)$$
$$= 2816.32\ kJ/\text{KG}.$$

Enthalpy of feed water entering the economiser

$$h_{f_1} = C_{pw} \times t_{e_1} = 4.1868 \times 30 = 125.60\ kJ/kg$$

Enthalpy of feed water leaving the economiser

$$h_{f_2} = C_{pw} \times t_{e_2} = 4.1868 \times 120 = 502.42\ kJ/kg$$

Heat of fuel utilized in economiser

$$Q_e = m_s(h_{f_2} - h_{f_1})$$
$$= 5600 \times (502.42 - 125.60) = 2110.192 \times 10^3\ kJ/\text{hr}$$

Heat of fuel utilized in boiler

$$Q_b = m_s(h_2 - h_{f_2})$$
$$= 5600 \times (2689.91 - 502.42) = 12249.944 \times 10^3\ kJ/\text{hr}$$

Heat of fuel utilized in superheater

$$Q_{sup} = m_s(h_{sup} - h_2)$$
$$= 5600(2816.32 - 2689.91) = 707.896 \times 10^3\ kJ/\text{hr}$$

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Total heat supplied to plant by fuel

$$= m_f.(CV) = 650 \times 31500 = 20475 \times 10^3 \ kJ/\text{hr}$$

(i) % or heat of fuel utilized in economiser

$$= \frac{Q_e}{Q_f} \times 100 = \frac{2110.192 \times 10^3}{20475 \times 10^3} \times 100 = \mathbf{10.3\%}$$ ***Ans***

% or heat of fuel utilized in boiler

$$= \frac{Q_b}{Q_f} \times 100 = \frac{12249.944 \times 10^3}{20475 \times 10^3} \times 100 = \mathbf{59.83\%}$$ ***Ans***

% or heat of fuel utilized in superheater

$$= \frac{Q_{\text{sup}}}{Q_f} \times 100 = \frac{707{:}896 \times 10^3}{20475 \times 10^3} \times 100 = \mathbf{3.45\%}$$ ***Ans***

% of Total heat of fuel utilized = (10.3 + 59.83 + 3.46) % = **73.59 %**

(ii) Overall efficiency of plant

$$\eta = \frac{\text{heat supplied to steam}}{\text{heat supplied by fuel}}$$

$$= \frac{m_s(h_{\text{sup}} - h_{f_1})}{m_f \cdot CV} \times 100 \quad \text{or} \quad \frac{\text{total heat utilized}}{\text{heat supplied}} \times 100$$

$$= \frac{5600(2816.32 - 125.60)}{650 \times 31500} \times 100 = \mathbf{73.59\ \%}$$ ***Ans***

(iii) Equivalent evaporation

$$m_{SE} = \frac{m_a(h_{\text{sup}} - h_{f_2})}{2257}$$ (Considering superheater is a part of boiler)

$$= \frac{5600}{650} \times \frac{(2816.32 - 502.42)}{2257}$$

$$= \mathbf{8.83\ kg\ of\ steam/kg\ of\ coal\ burnt}$$ ***Ans***

[Note : (1) Considering economiser and superheater both parts of boiler, then

$$m_{SE} = \frac{m_a(h_{\text{sup}} - h_{f_1})}{2257}$$

(2) Without considering superheater and economiser as parts of boiler, then

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$$m_{SE} = \frac{m_a (h_{\text{sup}} - h_{f_2})}{2257}]$$

Problem 2.2 *The following observations were made during the test of a boiler plant consisting of a 6 lancashire boilers and an economiser, mass of feed water per kg of coal = 9.5 kg, equivalent evaporation from and at 100°C = 9.8 kg/of coal burnt.*

Temperature of feed water to economiser = 15°C
Temperature of the flue gases entering economiser = 360° C
Temperature of air = 18° C
Mass of flue gases entering the ecomomiser = 18 kg/kg of coal.
Specific heat of flue gases = 1.046 kJ/kg °C
Calorific value of coal = 30,000 kJ/kg of coal.
Calculate : (i) efficiency of boiler alone. (ii) efficiency of the economiser alone (iii) efficiency of the whole plant.

Solution : Given data: $m_w = 9.5$ *kg/kg* of coal, $m_{SE} = 9.8$ *kg/kg* of coal,

$t_{w_1} = 15^{\circ}$ C, $t_{w_2} = 108^{\circ}$ C, $t_g = 360^{\circ}$ C, $t_a = 18^{\circ}$ C,

$m_g = 18$ *kg/kg* of coal, $C_{pg} = 1.046$ *kJ/kg* °C, CV = 30000 *kJ/kg*.

$$\text{(i) } m_{SE} = \frac{\text{Heat supplied to raise steam/kg of coal}}{2257}$$

$$\therefore 9.8 = \frac{Q_s}{2257}$$

$$\therefore Q_s = 22118.6 \text{ kJ/kg of coal}$$

$$\eta_{boiler\ alone} = \frac{\text{Heat supplied to raise steam/kg of coal}}{\text{Heat supplied by fuel / kg of fuel}} \times 100$$

$$= \frac{22118.6}{30,000} = \mathbf{73.72\%}$$ ***Ans***

(ii) Heat contained the flue gases entering economiser

$$= m_g \cdot C_{pg} \cdot (t_g - t_a)$$

$$= 18 \times 1.046 \times (360 - 18) = 6439.176 \text{ kJ/kg of coal}$$

Heat utilized in economiser (heat gain by feed water)

$$= m_w \cdot C_{pw} \cdot (t_{w_2} - t_{w_1})$$

$$= 9.5 \times 4.187 \times (108 - 15) = 3699.214 \text{ kJ/kg of coal}$$

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$$\eta_{economiser} = \frac{\text{heat utilized in economiser/kg of coal}}{\text{heat in fine gases entering in economiser/kg of coal}} \times 100$$

$$= \frac{3699.214}{6439.176} \times 100 = \mathbf{57.45\ \%} \qquad \textit{Ans}$$

(iii) Total heat utilized in the plant/*kg* of coal

= heat utilized in boiler + heat utilized in economiser

= 22118.6 + 3399.214

= 25517.814 *kJ/kg* of coal

Heat supplied to plant by fuel = 30,000 *kJ/kg* of coal

$$\therefore \eta_{whole\ plant} = \frac{25517.814}{30000} \times 100 = \mathbf{85.06\ \%} \qquad \textit{Ans}$$

Problem 2.3 *The following observations made during trial of water tube boiler.*

Steam pressure = 12 bar

Degree of super heat = 78°C

Temperature of feed water = 80°C

Water evaporated = 2800 kg/hr

Coal fired = 400 kg/hr

Ash = 45 kg/hr

Percentage of combustible in ash = 9.5 %

Moisture in coal = 4.5 %

Calorific value of coal = 31000 kJ/kg

Calorific value of carbon = 33850 kJ/kg

Specific heat of steam = 2.1 kJ/kg K

Calculate : (i) The efficiency of the boiler including superheater

(ii) The efficiency of boiler and furnace combined.

Solution : Given data : $t_{fw} = 80^{o}C$, $t_{sup} - t_s = 78^{o}C$, m_s = 2800 *kg*/hr,

m'_f = 400 *kg*/hr, CV of fuel = 31000 *kJ/kg*, C_{ps} = 2.1 *kJ/kg* K,

CV of carbon = 33850 *kJ/kg*.

From steam table, at 12 bar

h_f = 798.43 *kJ/kg*, h_g = 2782.7 *kJ/kg*, t_s = 187.96 °C

$$h_2 = h_{sup} = h_g + C_{ps}(t_{sup} - t_s)$$

$$= 2782.7 + 2.1(78) = 2946.5\ kJ/kg$$

$$h_1 = h_f = t_{fw} \times C_{pw} = 80 \times 4.187 = 334.96\ kg/kg$$

F- 6

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Actual mass of dry coal
= total coal supplied – mass of moisture in coal

$$= 400 - 4.5 \times \frac{400}{100}$$

$$m_f = 382 \ kg/\text{hr}$$

(i) Efficiency of boiler (including superheater)

$$\eta_b = \frac{m_s(h_2 - h_1)}{m_f \times CV} \times 100 = \frac{2800(2946.5 - 334.96)}{382 \times 31000} \times 100$$

$$= \frac{7312312}{11842000} \times 100 = \mathbf{61.75\ \%} \quad \textit{Ans}$$

Combustible in ash = mass of ash *kg*/hr × combustion in ash

$$= 45 \times \frac{9.5}{100} = 4.275\ kg$$

Actual heat supplied by fuel
= heat of dry coal – heat of combustible in the ash
= m_f × CV of fuel – 4.275 × C.V. of carbon
= 382 × 31000 – 4.275 × 33850
= 11697291.25 *kJ*/hr

(ii) Efficiency of boiler (including furnace)

$$\eta_b = \frac{m_s(h_2 - h_1)}{\text{actual heat supplied by fuel}} \times 100$$

$$= \frac{7312312}{11697291.25} \times 100 = \mathbf{62.51\ \%} \quad \textit{Ans}$$

Problem 2.4 *Following readings were taken from a steam power plant consisting of boiler, economizer and superheater :*
Steam pressure = 14 bar,
Mass of steam generated = 5000 kg/hr
Mass of coal used = 675 kg/hr
C.V. of fuel = 29800 kJ/kg of coal
Temperature of feed water entering economiser = 30° C
Temperature of feed water leaving the economiser = 130° C
Dryness fraction of steam leaving the boiler = 0.97
Temperature of steam leaving the superheater = 320° C
Determine (i) Overall efficiency of the plant. (ii) % of the available heat utilized in

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boiler, economiser and superheater respectively. *[G.U., EMCE, June-08]*

Solution : Given data : $m_s = 5000$ kg/hr. $m_f = 675$ kg/hr, $p = 14$ bar,

$CV = 29800$ kJ/kg, $t_{e_1} = 30^\circ$ C, $t_{e_2} = 130^\circ$C, $x = 0.97$, $T_{\text{sup}} = 320^\circ$ C

From steam table, at pressure p = 14 bar

$$h_f = 830.07 \text{ kJ/kg}, \; h_g = 2787.8 \text{ kJ/kg}, \; T_s = 195.04^\circ\text{C}$$

Enthalpy of steam leaving the boiler

$$h_2 = h_f + x(h_g - h_f)$$
$$= 830.07 + 0.95(2787.8 - 830.07)$$
$$= 2689.91 \text{ kJ/kg}$$

Enthalpy of steam leaving the superheater

$$h_{\text{sup}} = h_2 + C_{ps}(T_{\text{sup}} - T_s)$$
$$= 2689.91 + 2.3(320 - 195.04)$$
$$= 2977.318 \text{ kJ/kg}$$

Enthalpy of feed water entering the economiser

$$h_{f_1} = C_{pw} \times t_{e_1} = 4.187 \times 30 = 125.60 \text{ kJ/kg}$$

Enthalpy of feed water leaving the economiser

$$h_{f_2} = C_{pw} \times t_{e_2} = 4.187 \times 130 = 544.31 \text{ kJ/kg}$$

(i) Overall efficiency of plant

$$\eta = \frac{\text{heat supplied to steam}}{\text{heat supplied by fuel}} \times 100$$

$$= \frac{m_s(h_{\text{sup}} - h_{f_1})}{m_f \cdot CV} \times 100$$

$$= \frac{5000(2977.319 - 125.60)}{675 \times 29800} \times 100$$

$$\eta = \mathbf{70.88\%}$$ ***Ans***

(ii) Heat of fuel utilized in economiser

$$Q_e = m_s(h_{f_2} - h_{f_1})$$
$$= 5000(544.31 - 125.6)$$
$$= 2093.55 \times 10^3 \text{ kJ/hr}$$

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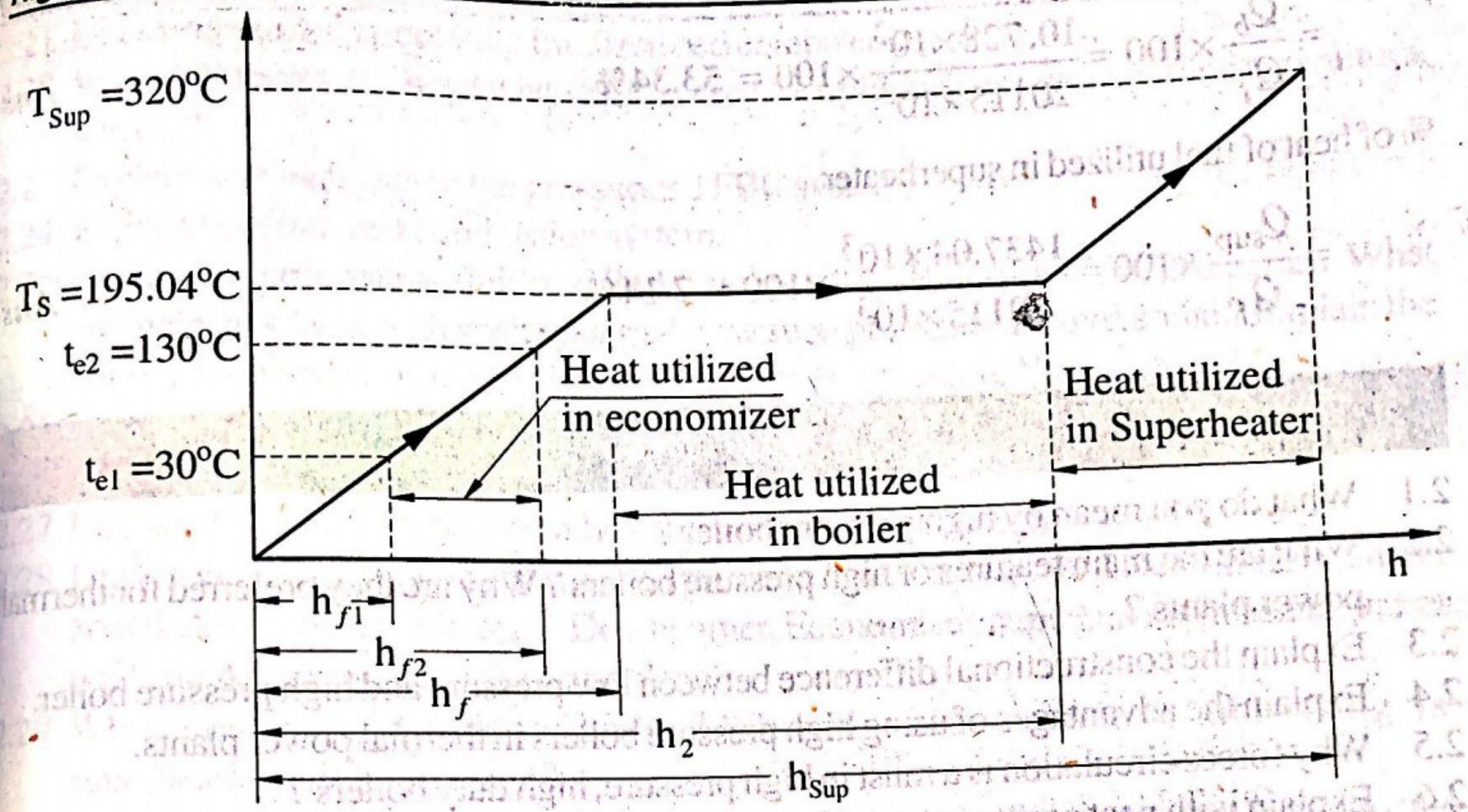


Fig. 2.35 T-h diagram for steam formation

Heat of fuel utilized in boiler

$$Q_b = m_s(h_2 - h_{f_2})$$

$$= 5000(2689.91 - 544.31)$$

$$= 10,728 \times 10^3 \text{ kJ/hr}$$

Heat of fuel utilized in superheater

$$Q_{sup} = m_s(h_{sup} - h_2)$$

$$= 5000(2977.318 - 2689.91)$$

$$= 1437.04 \times 10^3 \text{ kJ/hr}$$

Total heat supplied to plant by fuel

$$Q_f = m_f \times CV$$

$$= 675 \times 29800 = 20115 \times 10^3 \text{ kJ/hr.}$$

% of heat of fuel utilized in economiser

$$= \frac{Q_e}{Q_f} \times 100 = \frac{2093.55 \times 10^3}{20115 \times 10^3} \times 100 = \mathbf{10.40\%}$$ ***Ans***

% of heat of fuel utilized in boiler

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$$= \frac{Q_b}{Q_f} \times 100 = \frac{10,728 \times 10^3}{20115 \times 10^3} \times 100 = \mathbf{53.34\%} \quad \textit{Ans}$$

% of heat of fuel utilized in superheater

$$= \frac{Q_{sup}}{Q_f} \times 100 = \frac{1437.04 \times 10^3}{20115 \times 10^3} \times 100 = \mathbf{7.14\%} \quad \textit{Ans}$$

Exercises

2.1 What do you mean by high pressure boiler ?

2.2 What are the main features of high pressure boilers ? Why are they preferred for thermal power plants ?

2.3 Explain the constructional difference between low pressure and high pressure boiler.

2.4 Explain the advantages of using high pressure boilers in thermal power plants.

2.5 Why forces circulation is a must in high pressure, high duty boilers ?

2.6 Explain with neat construction and working of sketch LaMont boiler.

2.7 Explain with neat sketch construction and working of Benson boiler.

2.8 Draw and explain 'Schmidt-Hartmann Boiler', and discuss the merits and demerits over the high pressure boilers.

2.9 Explain a high pressure boiler which can carry higher salt concentration in feed water than any other type with neat sketch.

2.10 Explain supercharged boiler and supercritical boiler, giving examples of each type.

2.11 What is the difference between super critical and supercharged boilers ? Explain the construction, working of one supercritical boiler with diagram.

2.12 Explain construction, working and advantages of Loeffler boiler with neat sketch.

2.13 Explain Drumless boiler with a sketch.

2.14 "Part of superheated steam is only used in power generation in case of Loeffler boiler". Explain.

2.15 Explain in brief supercharged boilers are equipped with air compressor and gas turbine.

2.16 Draw neat line diagram of a high pressure boiler in which steam is generated by mixing the superheated steam with feed water. Discuss its relative merits and demerits.

2.17 What do you mean by supercritical boiler ?

2.18 Draw and explain supercharged boiler.

2.19 Write short notes on (a) Velox boiler and (b) Methods of desuper heating steam or super heat control.

2.20 What do you understand by fluidized bed combustion ? Explain working principle with a neat sketch. What are the major disadvantages of this system over conventional furnace boiler system ?

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2.21 Draw and explain circulating fluidized bed combustor.

2.22 What difficulties are faced when FBC is used for light fuels as municipal and agricultural wastes ?

2.23 Explain with neat sketch the pressurized FBC system.

2.24 Explain Fludised bed combustion system.

2.25 Explain the principle of fluidized Bed Combustion (FBC) systems with diagram. What are main problems with such system ? How such problems are overcome ? Explain the construction and working of classical/atmospheric pressure FBC system ?

2.26 Mention different advantages of fluidized bed system with respect to coal combustion. Why it is possible to burn any quality of coal in this system ?

2.27 Explain the main features and advantages of once through boiler.

2.28 Draw a neat sketch of a modern high pressure boiler and mark different heat transfer zoned. Also label Boiler drum, Downcorner, Economiser, superheater, Reheater, Water wall and Air preheater in the same sketch.

2.29 What are the major zones of boiler ? What are the considerations in locating the superheater and economizer ?

2.30 State and name and location of different types of heating surfaces used in a high pressure boilers. State their function.

2.31 Explain location of heating surfaces in water tube boilers.

2.32 Why superheating of steam is required ?

2.33 Why are superheaters and reheaters used in steam power plants ?

2.34 Why the heating of water is necessary before it is fed to the boiler ?

2.35 Why economizer is used in power plant ? Discuss the different type of economizers.

2.36 Explain types of superheater and state the various methods used for controlling the superheat temperature of steam.

2.37 With neat sketch, explain working of plain tube and gilled tube economizer.

2.38 Write shortnote on regenerative air preheater.

2.39 Why economizers are used in power plant ? Discuss different types of economizers.

2.40 Why economizers are located after the feed pumps ?

2.41 How does air preheating save fuel ?

2.42 Compare regenerative and recuperator types of air preheaters.

2.43 Why rotary regenerator type air preheater commonly used ?

2.44 Explain the function of air preheater in thermal power plant. With a neat sketch explain the construction and working of regenerative type of air-preheater.

2.45 Classify air preheaters and explain any one type of air preheater with neat sketch.

2.46 Describe with sketch the working of regenerative and recuperative types of air preheaters.

2.47 What are the causes of boiler corrosion and how it is prevented in practice ?

2.48 Explain corrosion of an economizer and its prevention.

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Objective Type Questions

1. Which of the following is not a high pressure boiler ?
   (a) Lancashire (b) Velox (c) La-Mant (d) Lo-effler
2. Indirect heating and evaporation of water is used in
   (a) La-Mont boiler (b) Benson boiler
   (c) Lo-effler boiler (d) Velox boiler
3. Once through boiler is named such because
   (a) air is sent through the same direction
   (b) steam is sent out only in one direction
   (c) there is no recirculation of water
   (d) flue gases pass only in one direction
4. Once through boilers will not have
   (a) drums, steam separators, pumps
   (b) drums, headers, steam separators
   (c) drums, headers, pumps
   (d) drums, headers, steam separators, pumps
5. Bension boilers having
   (a) three drums (b) one drum (c) no drum
6. Combustion in coal fired boilers usually takes place at
   (a) 1400 to 1700°C (b) 800 to 900°C
   (c) 200 to 300°C (d) none of these
7. In fluidized bed combustion (FBC) system the combustion of fuel takes place at about
   (a) 1400 to 1700°C (b) 800 to 800°C
   (c) 200 to 300°C (d) none of these
8. A super critical boiler is operates above the pressure
   (a) 221.05 bar (b) 100 bar (c) 80 bar (d) 50 bar
9. Which of the following statement is not correct regarding supercritical boilers.
   (a) No drum, less heat capacity of the generator and hence more stable and good response.
   (b) There is no two phase mixture.
   (c) Less corrosion and erosion problem compared to subcritical boilers.
   (d) None of the above
10. Which of the following is not a advantage of supercharged boiler.
   (a) Small heat capacity
   (b) Less heat transfer surfaces is required.
   (c) Rapid start of the boiler is possible.
   (d) It has a tight passage for gases.
11. Which of the following is/are fuel of FBC boiler.
   (a) bio mass (b) municipal waste
   (c) high ash coal (d) all of these

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12. The main function of economizer of a boiler plant is to
(a) reduce fuel consumption
(b) increase steam production
(c) increase steam pressure

13. In which device the temperature of steam is highest
(a) evaporator tubes (b) superheater
(c) economizer (d) steam/water drum

14. In a steam power plant heat from the flue gases is recovered in
(a) superheaters (b) de-superheater
(c) evaporator tubes (d) economizer and air preheater

15. Economizer in a steam power plant
(a) saves fuel consumption by 5-15%
(b) improves the boiler efficiency by 10-12%
(c) both of above

16. The advantage of reheating of steam in steam power plant
(a) it reduces the wears on the blades.
(b) it increases the efficiency of the turbine.
(c) it increases the work done through the turbine
(d) all of the above

17. Which one of the following modes of heat transfer would take place predominetly from boiler furnace to water walls ?
(a) radiation (b) convection (c) conduction

18. Which one of the following represents correct order in the path of flue gases ?
(a) superheater, economiser, evaporator tubes
(b) superheater, evaporator tubes, economizer
(c) evaporator tubes, superheater, economizer, air preheater.
(d) evaporator tubes, superheater, air preheater, economizer.

19. Which of the following method is not a method of superheat temperature control
(a) Desuperheating (b) Precondensing the steam
(c) Feed water heating (d) Tilting burner

Answers

1. (a)	**2.** (c)	**3.** (c)	**4.** (b)	**5.** (c)	**6.** (a)	**7.** (b)	**8.** (a)	**9.** (d)	
10.(d)	**11.** (d)	**12.** (a)	**13.** (b)	**14.** (d)	**15.** (c)	**16.** (d)	**17.** (a)	**18.** (c)	**19.** (c)

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GTU Study Materials ☑

Chapter 3

Coal and Ash Handling Systems

3.1 Introduction
3.2 Out-Plant handling of coal
3.3 Storage of coal
- *3.3.1 Dead storage or outdoor storage*
- *3.3.2 Live storage or active storage*

3.4 In-plant handling of coal
- *3.4.1 Coal delivery*
- *3.4.2 Unloading*
- *3.4.3 Preparation of coal*
- *3.4.4 Transfer of coal*
- *3.4.5 Inplant handling*
- *3.4.6 Weighing of coal*

3.5 Coal Burning System
3.6 Solid fuel firing
- *3.6.1 Hand firing*
- *3.6.2 Mechanical firing (stoker firing)*

3.7 Types of stoker firing
- *3.7.1 Principle of overfeed stokers*
- *3.7.2 Principle of underfeed stokers*

3.8 Pulverized fuel system
3.9 Pulverized fuel handling systems
- *3.9.1 Unit system*
- *3.9.2 Central (Bin) System*

3.10 Pulverizing mills
- *3.10.1 Ball mill*
- *3.10.2 Ball and race mill*
- *3.10.3 Bowl mill*
- *3.10.4 Hammer (impact) mill*

3.11 Pulverized coal burners
- *3.11.1 Long flame or U-flame or streamlined burner*
- *3.11.2 Short flame or turbulent burner*
- *3.11.3 Tangential burners*
- *3.11.4 Cyclone burner*

3.12 Oil burners
3.13 Necessity of ash disposal

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3.14 Ash handling systems
3.15 Mechanical ash-handling system
3.16 Hydraulic ash-handling system
3.16.1 Low-velocity system
3.16.2 High-velocity system
3.17 Pneumatic ash handling system
3.18 Steam jet ash handling system
3.19 Dust collection and its disposal
3.20 Mechanical dust collector
3.20.1 Gravitational separators
3.20.2 Cyclone separator
3.20.3 Wet type mechanical dust collector
30.21 Electrostatic precipitator (ESP)

3.1 Introduction

Thermal power plants use fossil fuels, i.e (1) solid fuel such as coal, (2) liquid fuel such as oil, and (3) gaseous fuel as gas. The choice of fuels, their preparation and feeding, and their methods of firing deserve special attention for a power plant engineer. Since our country is deficient in petroleum products, the main fuel used in the steam power plant is ***coal***.

The coal handling facility is the lifeline of a coal-fueled power plant. Modern plants have high coal demands because of the ever-increasing sizes of turbine generator units and the economic advantages of a single coal handling facility serving a multi-unit power plant. Thus, coal handling facilities have to become more flexible, more reliable, and capable of handling larger quantities of coal in less time than ever before.

In coal fired steam power plants, a large quantity of ash is produced. Ash produced is about 10 to 20% of the total coal burnt in the furnace. A 200 MW capacity power plant using Indian coals produces as large as 60,000 tons of ash per annum. Handling of ash is a problem because ash coming out of the furnace is too hot, it is dusty and irritating to handle and is accompanied by some poisonous gases. Therefore the ashes should be discharged and dumped at a sufficient distance from the power plant and locality.

3.2 Out-Plant handling of coal

The transportation of coal from coal mines to power plant is known as out-plant coal handling. With an increase in power plant capacity, the coal requirements per day are considerably large.

The mode of transporting coal from the mines to the power plant is determined by the location of the plant relative to the location of the mines and also by the available practical methods of transportation such as sea or river (waterways), railways, highways (road), and conveyors.

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(i) Transportation by waterways :

If the power plant is situated on the bank of a river or near the sea shore, it is often economical to transport coal in boats or barges. The coal is then unloaded mechanically by cranes or grab buckets. It is then placed in the storage yard or directly to the conveyor system to be carried to power plant. In U.S.A., many power plants located on the banks of the large river receive coal by barge as it is more economical than other modes of transport.

(ii) Transportation by Rail :

In India, the coal supply to power plants is mainly by railway as unfortunately river transportation is not available. This mode of transport plays very important role for power plants which are located interior. This mode of transport is still the most important means of transportation in common use. A railway sliding line is taken into the power station and coal is delivered either in the storage yard or close to the point of consumption. It can be unloaded by means of cranes, grab buckets or wagon tipplers. The capacity of the railway siding should be equal to three days requirement of the plant.

(iii) Transportation by Road :

The transportation of coal by road is possible for small and medium size plant only due to traffic restrictions and transportation cost. The great advantage of this system is that the coal can be carried directly into the power house upto the point of consumption.

(iv) Transportation by Ropeways :

The transportation of coal by ropeways is particularly used when the distance between the mine and power plant is less than 10 kilometers. This mode of transportation is very efficient. It supplies the coal continuously and free from worker's strike.

(v) Transportation of coal by pipeline :

The power plants are far away from the coal mines and the existing railway facilities are not sufficient to cope with the increasing demands in the most of the countries.

The transportation of coal by pipeline is considered most speedy and efficient method among all available. In this mode of coal transportation, coal slurries from remote mine are supplied to power plant by pipeline. The following are the advantages offered by this method:

- More economical than other modes of transport when dealing with large volume of coal over long distances.
- This system is continuous as it remains unaffected by the variations of climate and weather.
- It provides simplicity in installation and increased safety in operation.
- It is capable of transporting very large quantities of coal.
- It offers high degree of reliability and safety as the moving machines are limited to the stationary pumping and boosting station.
- Noise and dust problem is considerably reduced.
- Manpower requirement is low.

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- **Disadvantages :**
- It requires high capital and operating cost.
- It requires large quantity of water as 1 kg of coal requires 1 kg of water.
- Power plants must be able to use coal with added surface moisture.

3.3 Storage of coal

The main purpose of coal storage is

- To store the coal for a period of 30 to 90 days, therefore the plant is not required to be shut down due to failure of normal supplies of coal.
- To permit choice of the date of purchase allowing the management to take advantage of seasonal market conditions in prices of coal.

Storage of coal is primarily a matter of protection against coal strikes, failure of transportation system and general coal shortages.

The large storage of coal is undesirable, because it costs more as there is risk of spontaneous combustion, interest on capital cost of coal, possibility of loss and deterioration during storage, handling costs required by storage and reclamation, cost of area required and many others. A storage of 10% of the annual consumption is sufficient in most of the cases, although the factors such as nearness to coal fields, transportation facilities and weathering effects on coal must also be considered.

The coal storage area includes both dead storage or outdoor storage and live storage or active storage.

3.3.1 Dead storage or outdoor storage

In this storage, *the coal received at the power plant is stored in dead storage in the form of piles laid directly on the ground.* The coal stored has the tendency to combine with oxygen of air and during this process coal loss some of its heating value and ignition quality. Due to oxidation the coal may ignite spontaneously. This can be avoided by storing coal in following ways :

(i) Stocking the coal in heaps or piles :

Generally concrete floored area is used to prevent the flow of air from the bottom for stocking the coal. Coal is stored in the form of heaps or piles upto a height of 10 to 12 m directly on the concrete ground. During storage of coal in heaps, the coal should be compacted in layer of 15 to 30 cm in thickness. This effectively prevents the air circulation in the interior of the pile. The pile top should be given a gentle slope in the direction in which the rain may be drained off.

Second method of removing the heat of oxidation is the air is, allowed to move through the layers evenly so that air may remove the heat of reaction and avoid burning.

The sealing of stored pile is desirable in order to avoid the oxidation of coal after packing an air tight layer of coal. Asphalt, fine coal dust, and bituminous coating are the materials commonly used for this purpose.

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(ii) Under water storage :

The possibility of slow oxidation and spontaneous combustion can be completely eliminated by storing the coal under water.

The following points should be kept in mind during selecting the site for storage and piling:

(1) Coal should be stored on solid ground free from water.
(2) The artificial drainage in the storage area should be provided if well drained area is not available.
(3) The coal handling cost should be minimum.
(4) Fire fighting equipment must be available near the storage.
(5) The piles should be built-up as far as possible compact.
(6) The coal storage area should be cleared of all foreign matter such as wood, paper, etc. having a low ignition temperature.

3.3.2 Live storage or active storage :

Live coal storage implies the reclaiming and combustion of coal that has been stored for only a relatively short time, usually less than a week. Coal from a live coal storage pile is usually supplied to combustion equipment without the use of mobile equipment. The coal is usually stored in the vertical cylinder bunkers or coal bins or silo. Coal from silos is transferred to the boiler grate.

Live coal storage bunkers are normally constructed with a diamond - shaped cross-section storage area. The storage bunkers are made of steel or reinforced concrete to store the live coal as shown in Fig. 3.1.

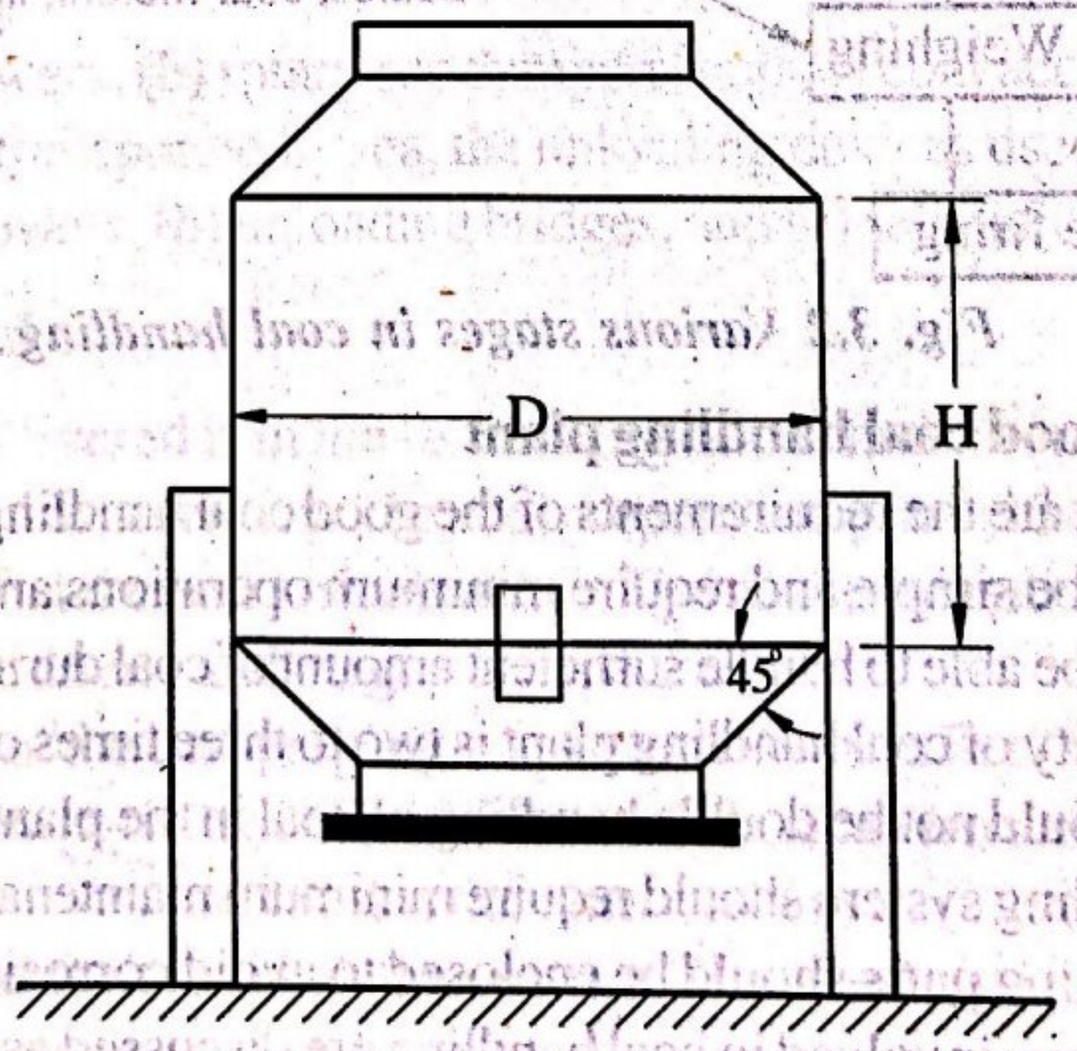


Fig. 3.1 Cylindrical bunker

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3.4 In-plant handling of coal or Stages of coal handling [Jan. '13]

Whatever may be the method of coal transporting but the coal has to be taken to the boiler stokers or the coal preparation plant in case of pulverized fuel firing. In large power stations, some mechanical handling system must be introduced to the plant for easy and smooth operation and better control. The various stages in coal handling are shown in Fig. 3.2.

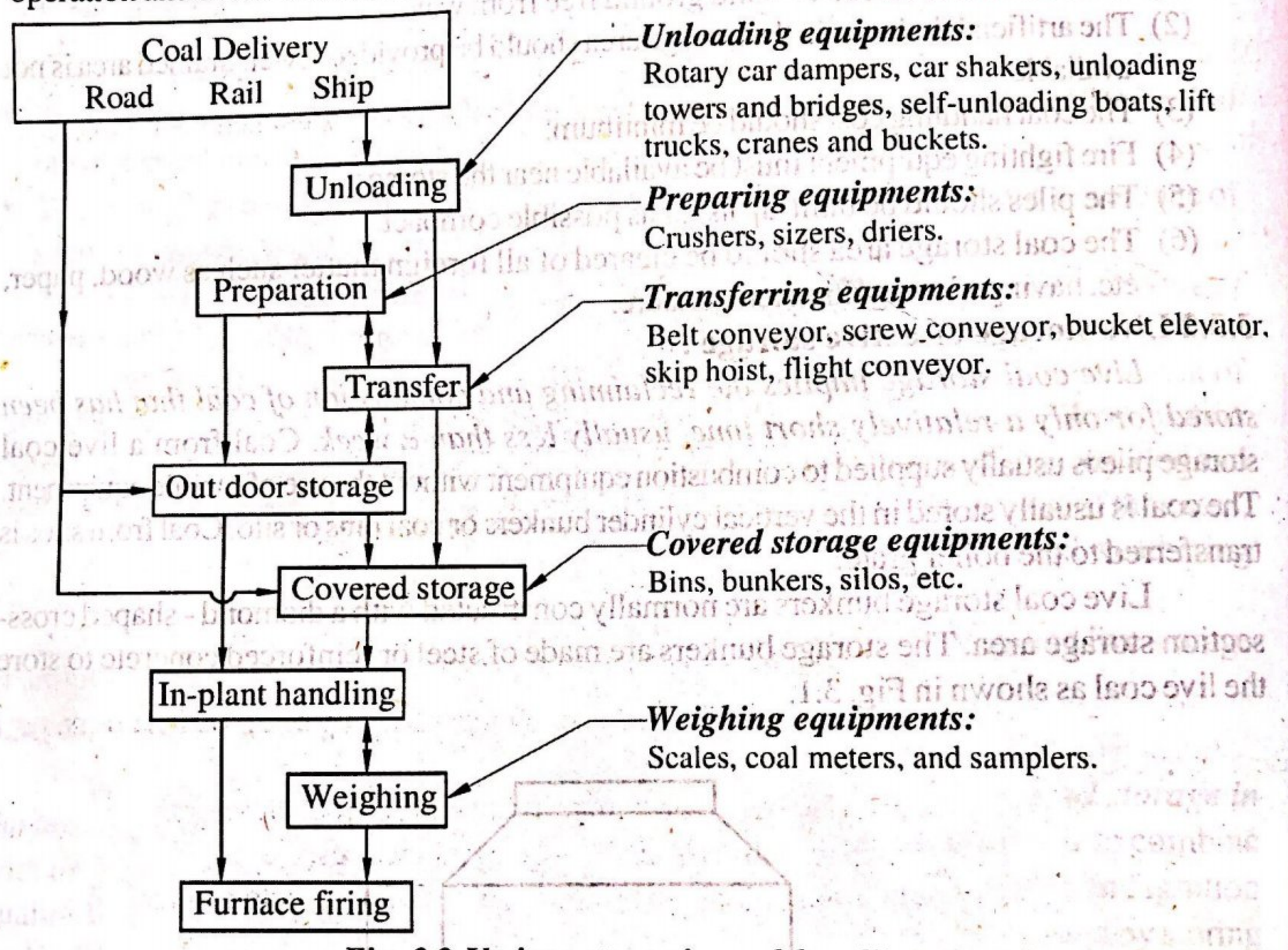


Fig. 3.2 Various stages in coal handling

- **Requirement of good coal handling plant**

The following are the requirements of the good coal handling system :

(1) It should be simple and require minimum operations and transport.

(2) It should be able to handle sufficient amount of coal during peak hours. Generally, the capacity of coal handling plant is two to three times of its normal requirement.

(3) There should not be double handling of coal in the plant.

(4) The handling system should require minimum maintenance and repair.

(5) The working parts should be enclosed to avoid corrosion and abrasion.

The various steps involved in coal handling are discussed as follows.

3.4.1 Coal delivery

The coal from supply points is delivered by ships or boats to power stations situated near to sea or river, whereas coal is supplied by rail or trucks to the power stations which are

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situated away from sea or river. The transportation of coal by trucks is used if the railway facilities are not available.

3.4.2 Unloading

The type of equipment to be used for unloading the coal received at the power plant depends upon the type of out-plant handling mode as road, rail, or ship. Road transportation of coal by trucks is often more economical for small and medium capacity plant within 30 to 50 kilometers of coal mines. If coal is delivered by trucks, there is no need of unloading device as the trucks may dump the coal to the outdoor storage. Coal is easily handled if the lift trucks with scoop are used as shown in Fig. 3.3.

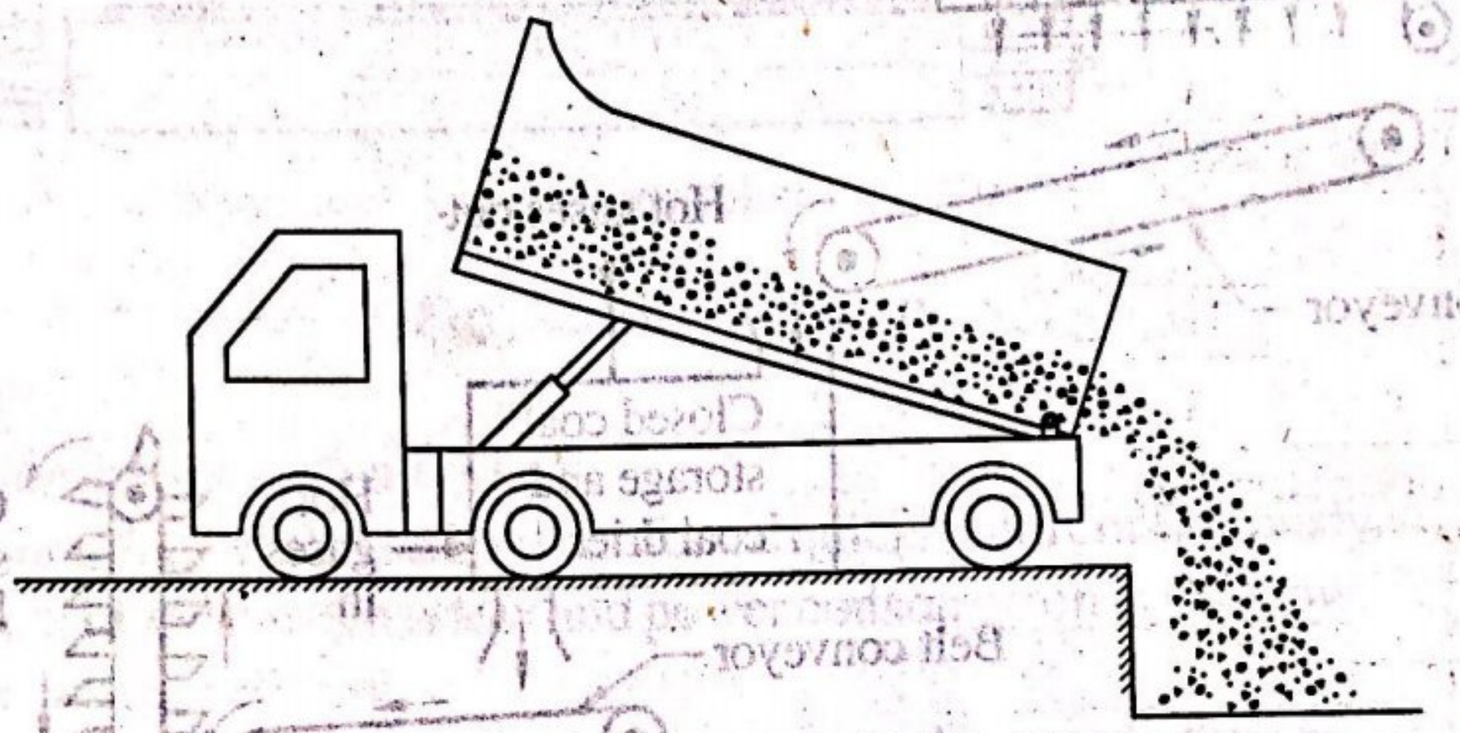


Fig. 3.3 Lift truck with scoop

When the coal is transported by railway, the following unloading equipment is normally used for speedy unloading :

(a) car towers, (b) rotary car dumpers, and (c) coal accelerators.

When the coal is transported by sea, the unloading devices used are :

(a) coal towers, (b) unloading bridges, and (c) self-unloading boats.

3.4.3 Preparation of coal

When the coal delivered is in the form of big lumps and it is not of proper size, the preparation (sizing) of coal is necessary. The coal preparation plant either can be located near the coal receiving point or at the point of actual use. The coal preparation can be achieved by crushers, breakers, sizers, driers, and magnetic separators.

The arrangement of the coal preparation plant is shown in Fig. 3.4. The coal crusher is used for crushing coal of required size. Part of the coal obtained from mines does not require sizing and is bypassed. The crushed coal is then passed over the vibrating separator which removes unsized coal and feeds back to the crusher. The sized coal is further passed to the drier to remove the moisture from coal. For removing the moisture from coal, hot flue gases are passed through the coal storage in closed spaces. The magnetic separators are used for removing the iron particles and scrap from the coal before supplying to the storage hopper. The separator is usually a magnetized pulley over which the loaded coal belt is run. The iron

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particles drop off into a reject chute as the belt leaves the pulley. Bucket elevator is used for transferring coal from the magnetic separator to the storage hopper.

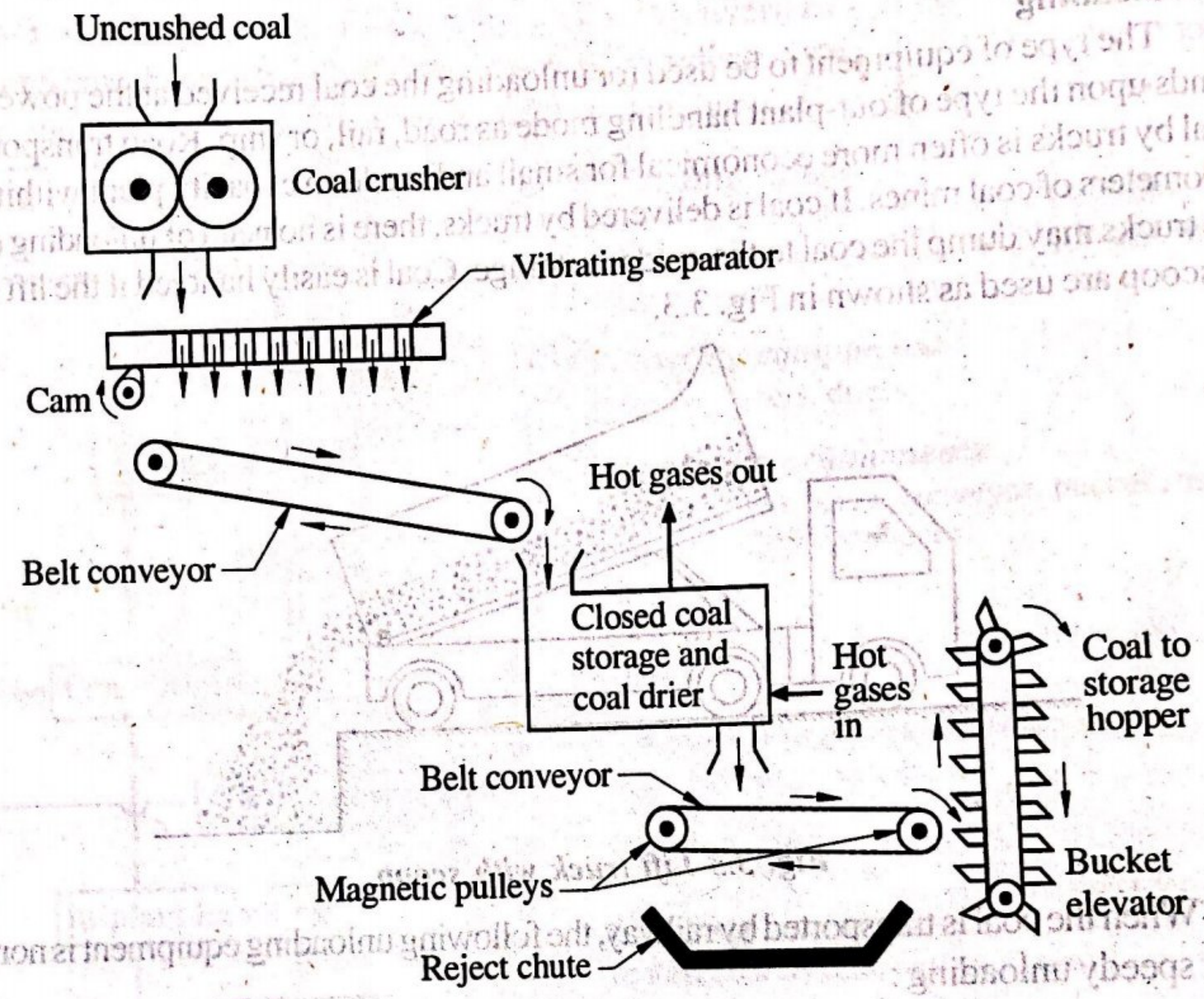


Fig. 3.4 Coal preparation plant

3.4.4 Transfer of coal

Coal is required to transfer from unloading point to the storage site. After preparation coal is transferred to the storage by means of the following systems.

(a) Belt -conveyors
(b) screw conveyors
(c) Bucket elevators
(d) Grab bucket elevators
(e) skip hoists, and
(f) Flight conveyors

(a) Belt conveyors :

Fig. 3.5 shows a belt conveyor. It consists of an endless belt moving over a pair of pulleys or end drums and supported by a series of rollers known as idlers provided at regular intervals. The belt is made up of rubber or canvas. The return rollers are plain rollers and are spaced wide apart which support the empty belt. Belt conveyor is very suitable means of transporting large quantities of coal over large distances. It is used in medium and large power plants. The inclination at which coal can be successfully elevated by belt conveyor is about 20° to the horizontal. An average speed of belt conveyors varies from 1 m/s to 1.8 m/s. It can transport a coal from 50 tonnes/hr to 100 tonnes/hr continuously.

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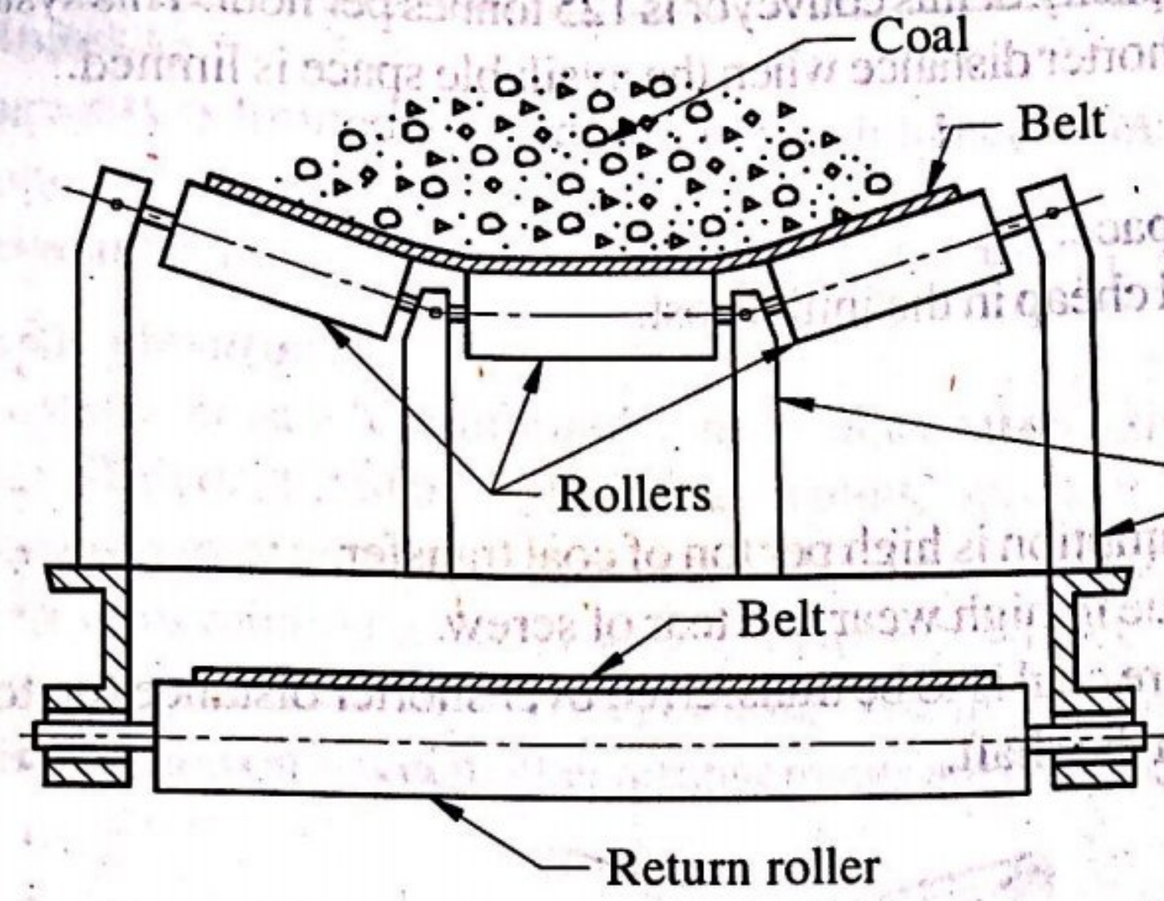


Fig. 3.5 Belt conveyor

- **Advantages :**
  (1) Its operation is smooth and clean.
  (2) Large quantities of coal can be transferred quickly and continuously.
  (3) Initial cost of the system is low and power consumption is also low.
  (4) Maintenance cost is low.
  (5) The coal transfer rate can be varied by varying the speed of the belt.
- **Disadvantages :**
  (1) It cannot be used to transfer the coal at greater heights as its inclination is limited to 20°.
  (2) It is not suitable for short distances.

(b) Screw conveyors :

Screw conveyor is shown in Fig. 3.6. It consists of an endless helicoid screw fitted to a shaft. The screw while rotating in a trough transfers the coal from supply end to the discharge end. The driving mechanism is connected to one end of the shaft and the other end of the shaft is supported in an enclosed ball bearing. Rotation of screw varies between 70-125 r.p.m. The diameter of screw varies from 15 cm to 50 cm.

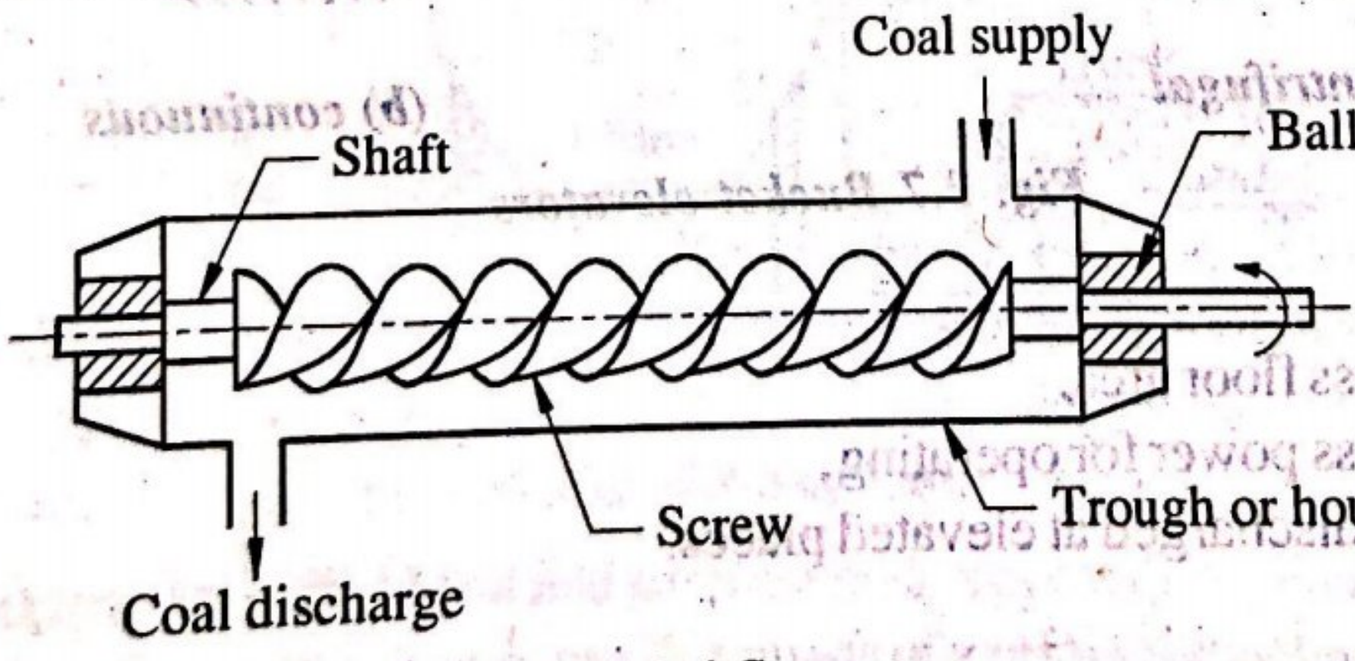


Fig. 3.6 Screw conveyor

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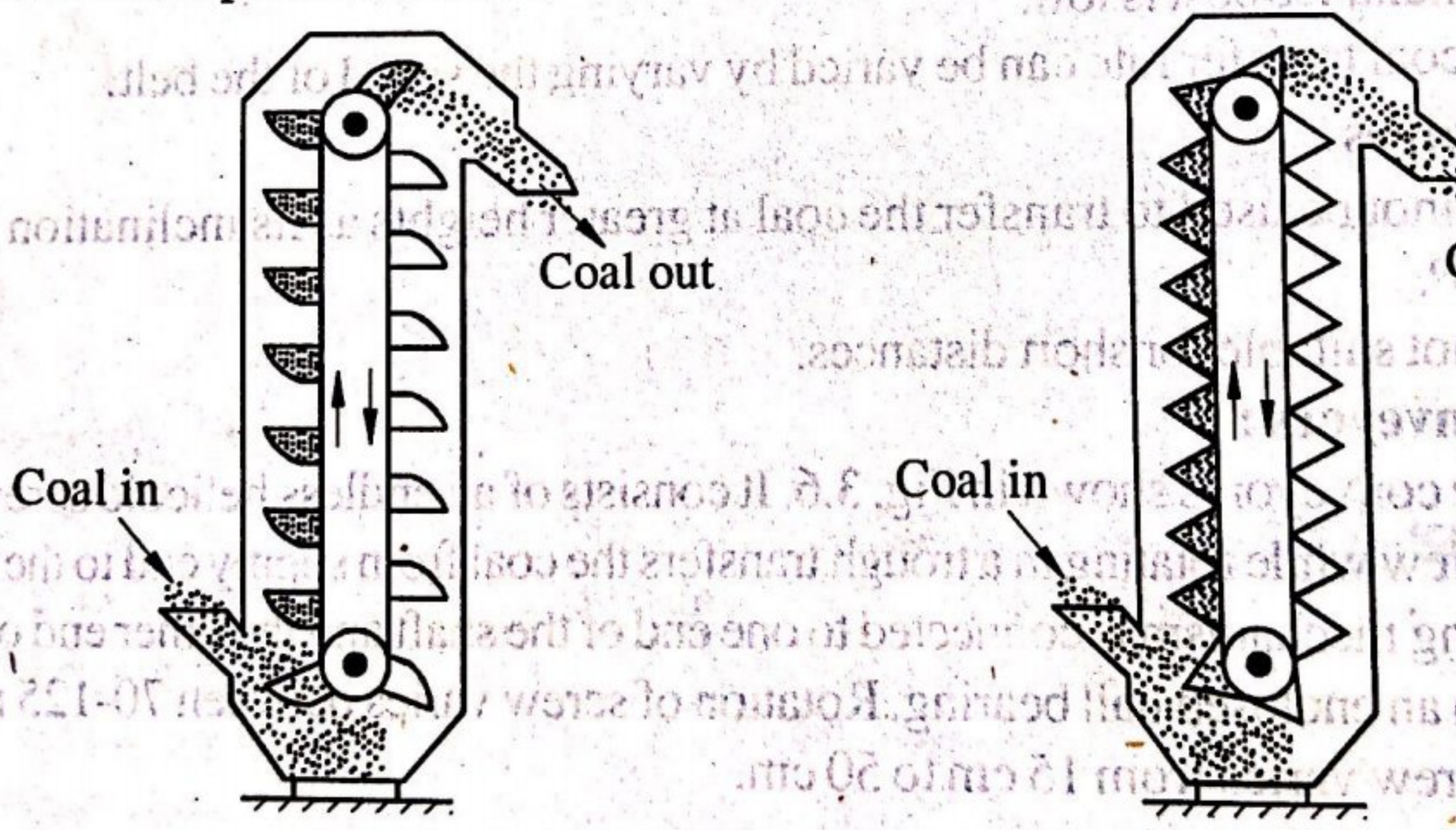
The maximum capacity of this conveyor is 125 tonnes per hour. This system is suitable for transfer of coal over shorter distance when the available space is limited.

- **Advantages :**
  - (1) It requires less space.
  - (2) It is compact and cheap in the initial cost.
  - (3) It is dust tight.

- **Disadvantages :**
  - (1) The power consumption is high per ton of coal transfer.
  - (2) It's life is short due to high wear and tear of screw.
  - (3) It is suitable, where coal is to be transferred over shorter distance (up to 30 m) due to torsional strain on the shaft.

(c) Bucket elevators :

It consists of buckets fixed to a chain as shown in Fig. 3.7(a). The chain moves over two wheels. The coal is carried by the buckets form bottom and discharged at the top. This conveyor is extensively used for vertical lifts. The material elevated in buckets is discharged either by centrifugal force at the top of the elevator or by drawing back the buckets on the discharge side. Anther continuous type bucket elevator is shown in Fig. 3.7(b), it carries more amount of coal compared with first.



(a) centrifugal *(b) continuous*

Fig. 3.7 Bucket elevators

- **Advantages :**
  - (1) It requires less floor area.
  - (2) It requires less power for operating.
  - (3) Coal can be discharged at elevated places.

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- **Disadvantages :**
  (1) Its capacity is limited to 70 tonnes per hour hence not suitable for large capacity stations.
  (2) The maximum height of the elevator is limited to 30.5 m.

(d) Grab bucket elevator :

It is a form of hoist which lifts and transfers the coal on a single rail or track from one point to another. The coal lifted by grab buckets is transferred to overhead bunker or storage. The grab bucket conveyor with crane is shown in Fig. 3.8. This system requires less power for operation and requires minimum maintenance. A 2 to 3 m^3 bucket operating over a distance of 60 m transfers nearly 100 tonnes of coal per hour. The initial cost of this machine is high, therefore use is justified only when other arrangements are not possible.

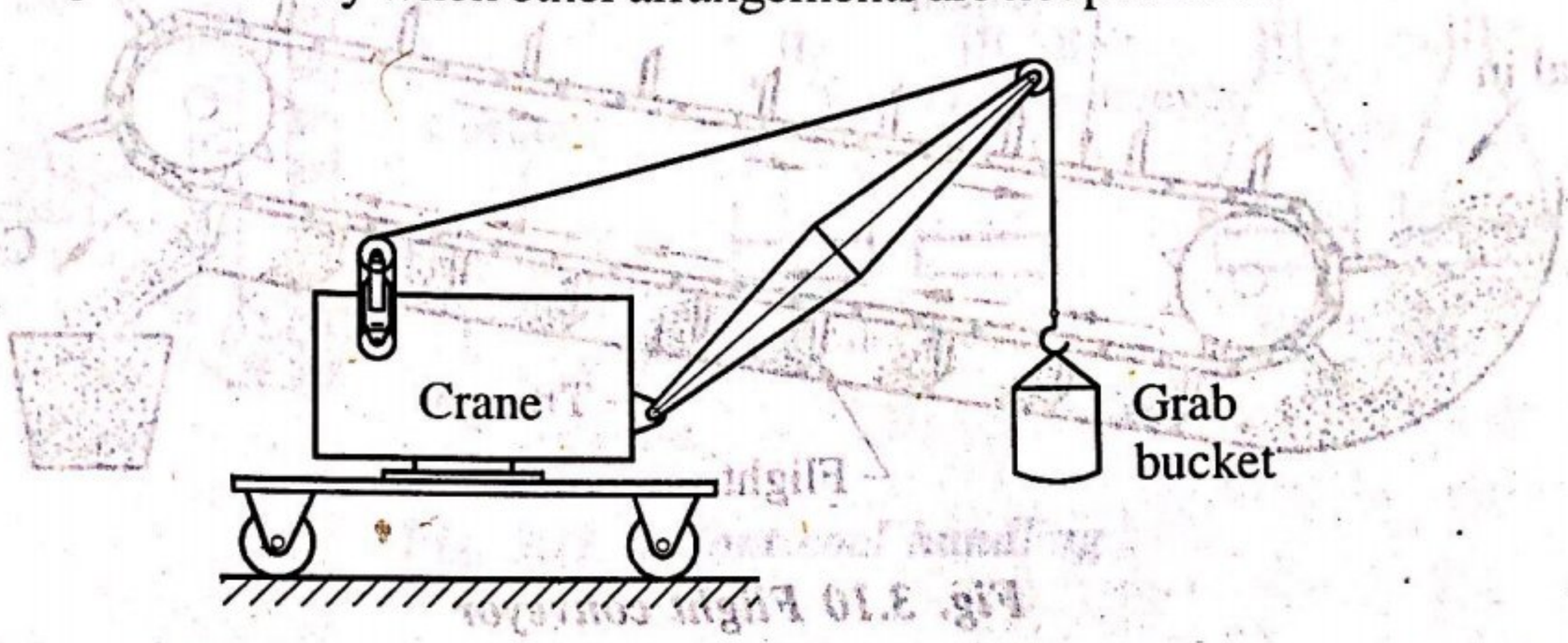


Fig. 3.8 Grab bucket elevator

(e) Skip hoist :

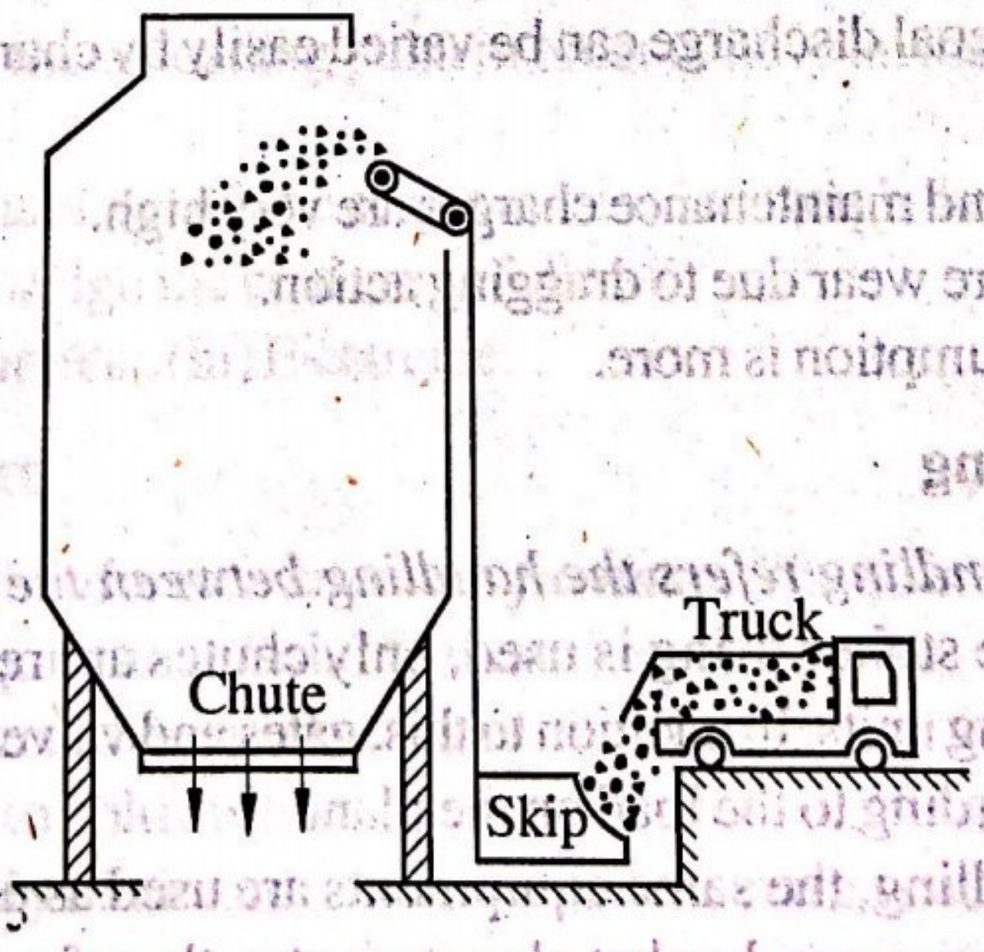


Fig. 3.9 Skip hoist

The skip hoist is the oldest and simplest means of elevating coal or ash. It is used for high lifts and handling is non-continuous. It consists of a vertical or inclined hoistway, a bucket

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or a car guided by a frame, and a cable for hoisting the bucket. The bucket is held in up right position. The operation and maintenance cost of this arrangement is low. Fig. 3.9 shows the skip hoist.

(f) Flight conveyor or scraper :

Flight conveyor consists of one or two strands of chain to which steel scraper or flights are attached. The scraper scrap the coal through a trough having identical shape. This coal is discharged in the bottom of trough. This conveyor is generally used for transfer of coal when filling of number of storage bins situated under the conveyor is required. The arrangement of flight conveyor is shown in Fig. 3.10.

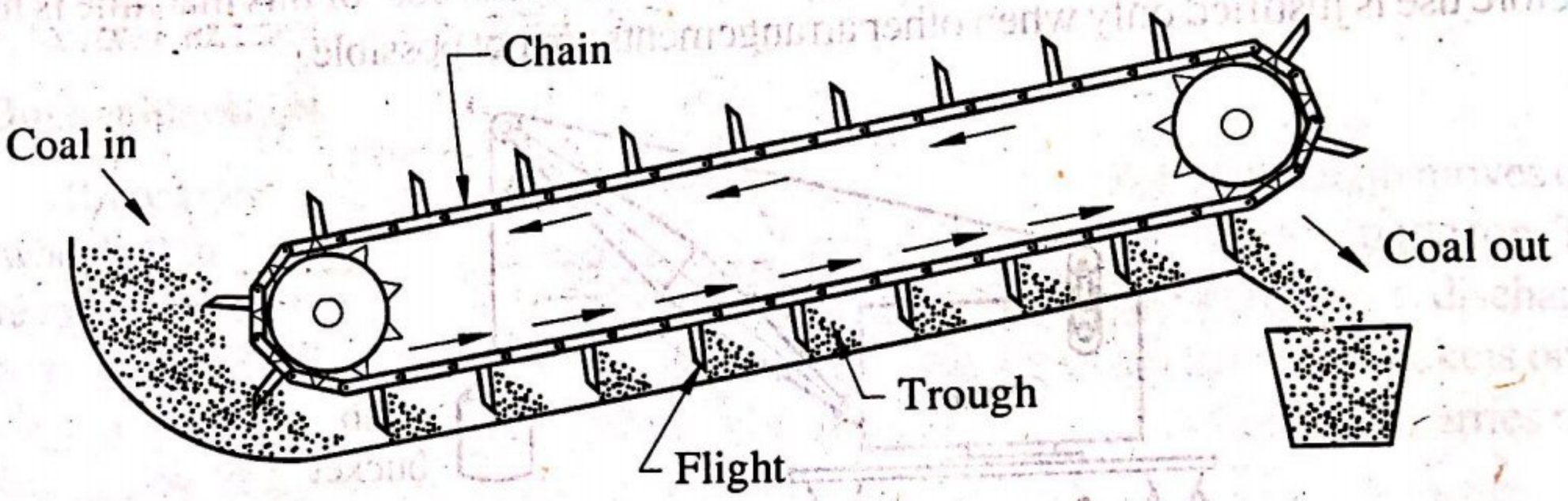


Fig. 3.10 Flight conveyor

- **Advantages :**
  - (1) It is low in first cost.
  - (2) It can be used to transfer coal as well as ash.
  - (3) The rate of coal discharge can be varied easily by changing the speed of conveyor.
- **Disadvantages :**
  - (1) The repair and maintenance charges are very high.
  - (2) There is more wear due to dragging action.
  - (3) Power consumption is more.

3.4.5 Inplant handling

Inplant coal handling refers the handling between the final storage and the firing equipment. If simple stoker firing is used, only chutes are required to feed the coal from storage bunker to firing units. In addition to this, gates and valves are included in the system to control the flow according to the load on the plant.

For inplant handling, the same equipments are used as used for coal transfer like belt conveyors, screw conveyors, bucket elevators etc., therefore it is not necessary to repeat them again. A typical inplant coal handling system is shown in Fig. 3.11.

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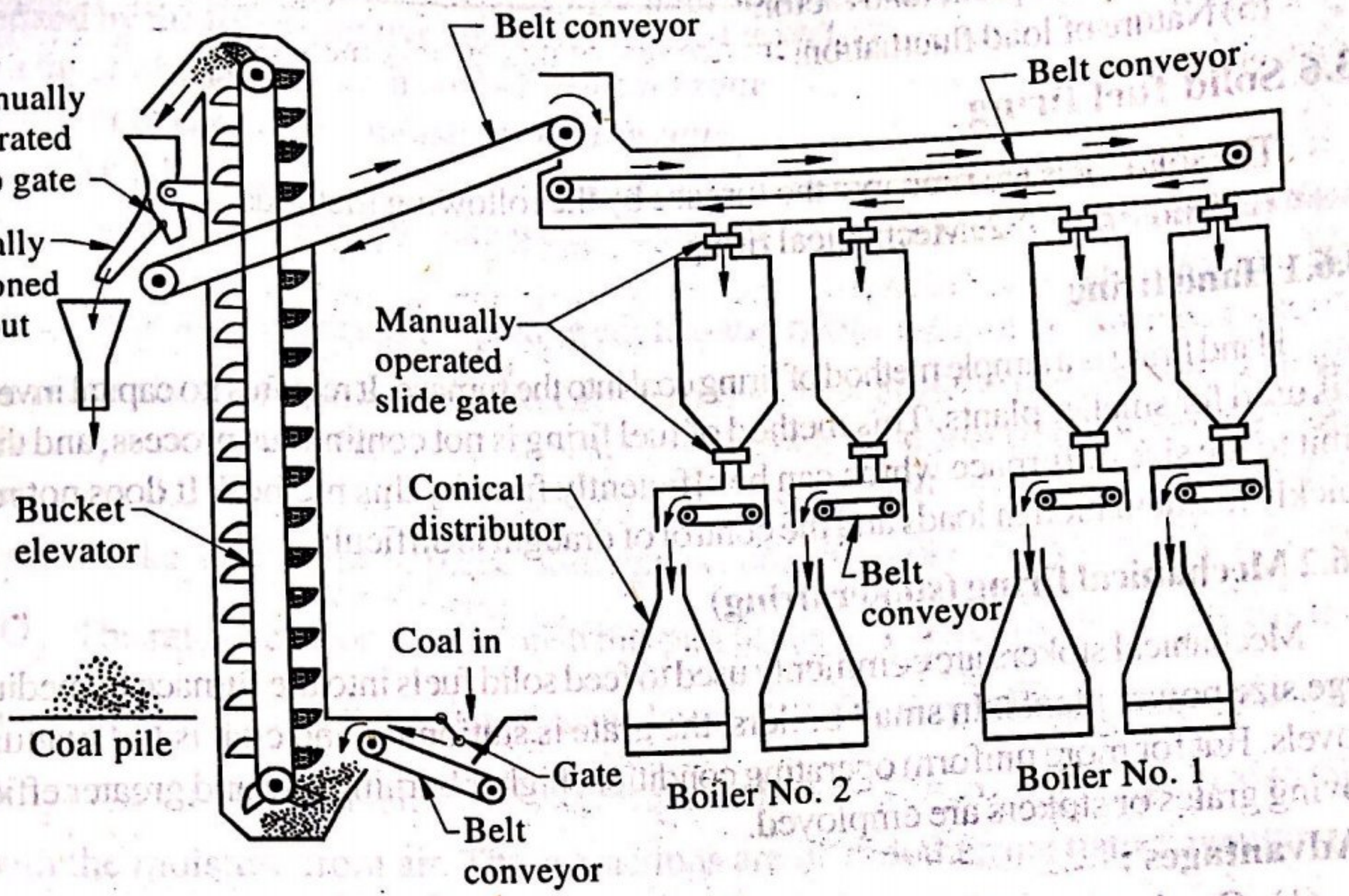


Fig. 3.11 Inplant coal handling

3.4.6 Weighing of coal

The coal before supplying to individual boilers, it is required to weigh in order to know whether the proper quantity of coal is supplied or not as per the load on the plant. It is also required to weigh the coal at the unloading point in order to have an idea of the total quantity of coal delivered at the site.

Weigh lorries, hoppers and automatic scales are used to weigh the quantity of coal. The commonly used methods to weigh the coal are as follows :

(i) Mechanical, (ii) Pneumatic, (iii) Electronic.

3.5 Coal Burning System

Fuel is burnt in a confined space know as ***furnace.*** The furnace provides supports and enclosure for burning equipment. Solid fuels such as coal, coke, wood, etc. are burnt by means of stokers whereas burners are used to burn pulverized coal and liquid fuels. Solid fuels require a grate in the furnace to hold the bed of fuel. In case of coal fired plants, the factors like fuel size, rate of firing, supply of air and moisture content in the coal affect the combustion process.

The selection of fuel firing methods depends upon the following factors :

(1) The characteristics of the available coal

(2) Capacity of the plant.

(3) Efficiency and reliability of the combustion equipments.

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(4) The power plant load factor.
(5) Nature of load fluctuation.

3.6 Solid fuel firing

The solid fuels are fired into the furnace by the following methods.
1. Hand firing 2. Mechanical firing

3.6.1 Hand firing

Hand firing is a simple method of firing coal into the furnace. It requires no capital investment. It is used for smaller plants. This method of fuel firing is not continuous process, and there is a limit to the size of furnace which can be efficiently fired by this method. It does not respond quickly to fluctuation in loads and the control of draught is difficult.

3.6.2 Mechanical firing (stoker firing)

Mechanical stokers are commonly used to feed solid fuels into the furnace in medium and large size power plants. In small boilers, the grate is stationary and coal is fed manually by shovels. But for more uniform operating condition, higher burning rate and greater efficiency, moving grates or stokers are employed.

- **Advantages :**
  (1) Combustion efficiency is high.
  (2) Coal preparation is not required.
  (3) Low grade coal can be used.
  (4) Firing system is reliable.
  (5) Less labour is required.
  (6) Smoke produced is less.
  (7) Free from danger of explosion.
- **Disadvantages :**
  (1) Construction is complicated and initial cost is high.
  (2) Loss of coal in the form of riddling through the grates.
  (3) High maintenance and repair cost due to high furnace temperatures.
  (4) There is excessive wear of moving parts due to abrasive action of coal.
  (5) Sudden variations in the steam demand cannot be met to the same degree.

3.7 Types of stoker firing

A grate is used at furnace bottom to hold a bed of fuel. There are two ways of feeding coal on to the grate : (a) overfeeding, and (b) underfeeding. Therefore automatic stokers may be classified as overfeed stokers and underfeed stokers.

3.7.1 Principle of overfeed stokers [Jan. '13, June '11]

In this case, the coal is fed into the grate above the point of air admission. The principle of overfeed stokers is shown in Fig. 3.12. It receives fresh coal on its top surface and is

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characterized by the following five zones from top to the bottom.

(a) a layer of fresh or green coal - ***fresh coal zone***.

(b) a layer of coal losing moisture - ***drying zone.***

(c) a coking layer of coal losing its volatile content - ***distillation zone.***

(d) a layer of incandescent coke where the fixed carbon is consumed - ***combustion zone.***

(e) a layer of ash progressively getting cooler - ***ash zone.***

The pressurised air coming from Forced Draft (F.D.) fan enters under the bottom of the grate. The air passing through the grate is heated by absorbing the heat from the ash and grate itself, whereas the ash and grate are cooled. The hot air then passes through a bed of incandescent coke. As the hot air passes through incandescent coke, the O_2 reacts with C to form CO_2. The rate of carbon-oxidation in this part of fuel bed depends entirely on the rate of air supply. In the event of thick incandescent layer, the CO_2 may partly or fully reduced to CO ($CO_2 + C = 2CO$). Further, water gas reaction ($H_2O + C = H_2 + CO$) may also take place with the moisture from air. These reactions are of endothermic nature resulting in considerable temperature drop of bed and gas stream.

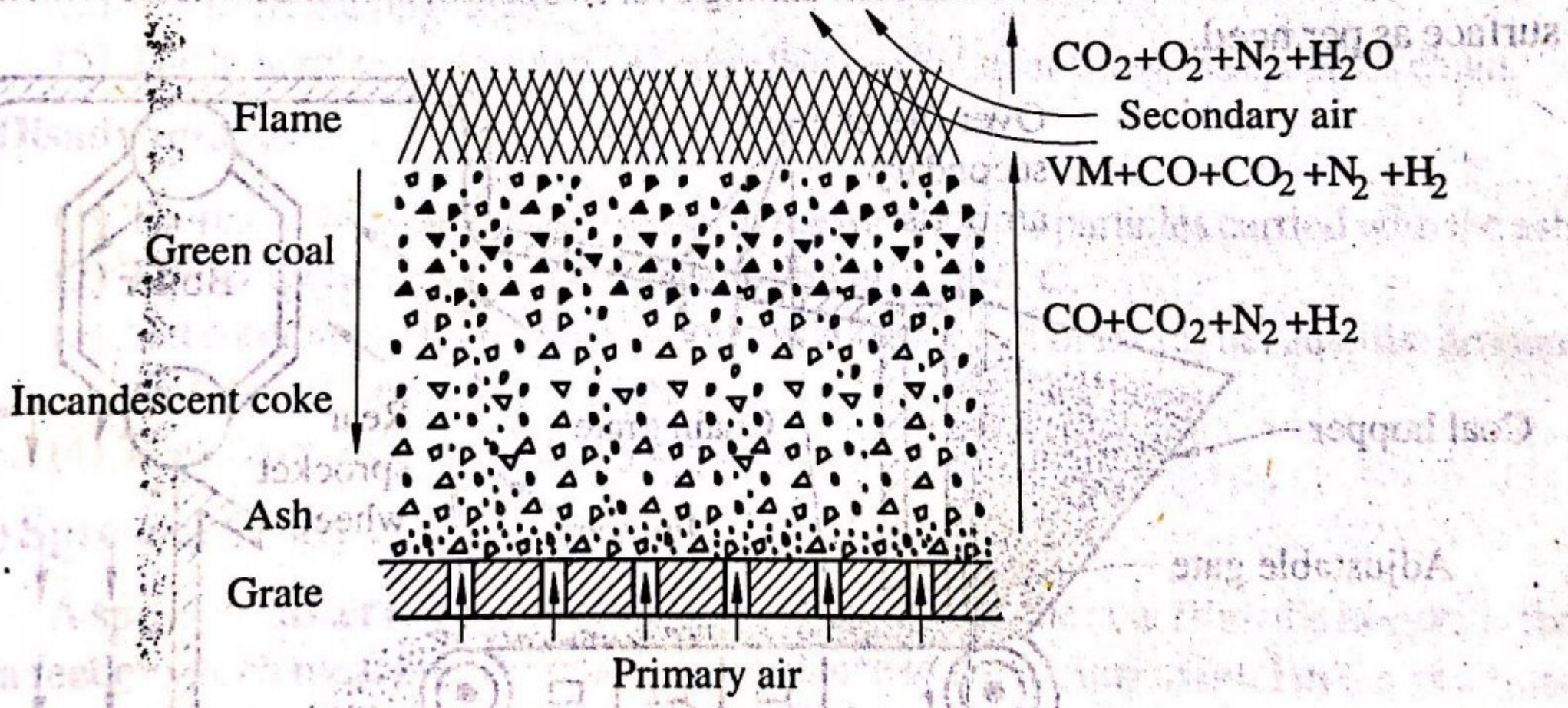


Fig. 3.12 Principle of overfeed stoker

As the hot gas stream passes through the distillation zone where the volatile matter is added and then through the drying zone where the moisture is picked up and finally comes out from the fuel bed containing N_2, CO_2, CO, H_2, combustible volatile matter and water vapour (H_2O). Then additional air known as ***secondary air*** is supplied at right angles above the fresh coal zone to burn the combustible gases (volatile matter + CO + H_2). The secondary air

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is supplied at a very high speed to create turbulence which is required for complete combustion of unburnt gases. The combustion gases entering the boiler consist of N_2, CO_2, O_2 and H_2O and also CO if the combustion is not complete.

During combustion, the incandescent coal continuously losses its carbon and only the ash remains which falls through the grate into the ash pit.

- **Types of overfeed stokers**

The overfeed stokers may be of the following types :
(1) Travelling grate stoker,
(2) Spreader stoker

(1) Travelling grate stoker : **[Dec. 10]**

The travelling stoker may be (i) chain grate type, or (ii) bar grate type. These two differ only in the details of grate construction. The travelling grate stoker is also known as ***conveyor grate stoker***.

In the chain grate stoker, the grate is made of a series of cast iron links connected by bars or pins to form an endless chain.

In the bar grate type stoker, the grate surface consists of a series of cast-iron bars joined together by links to form an endless belt running over two sets of sprocket wheels with a wide surface as per need.

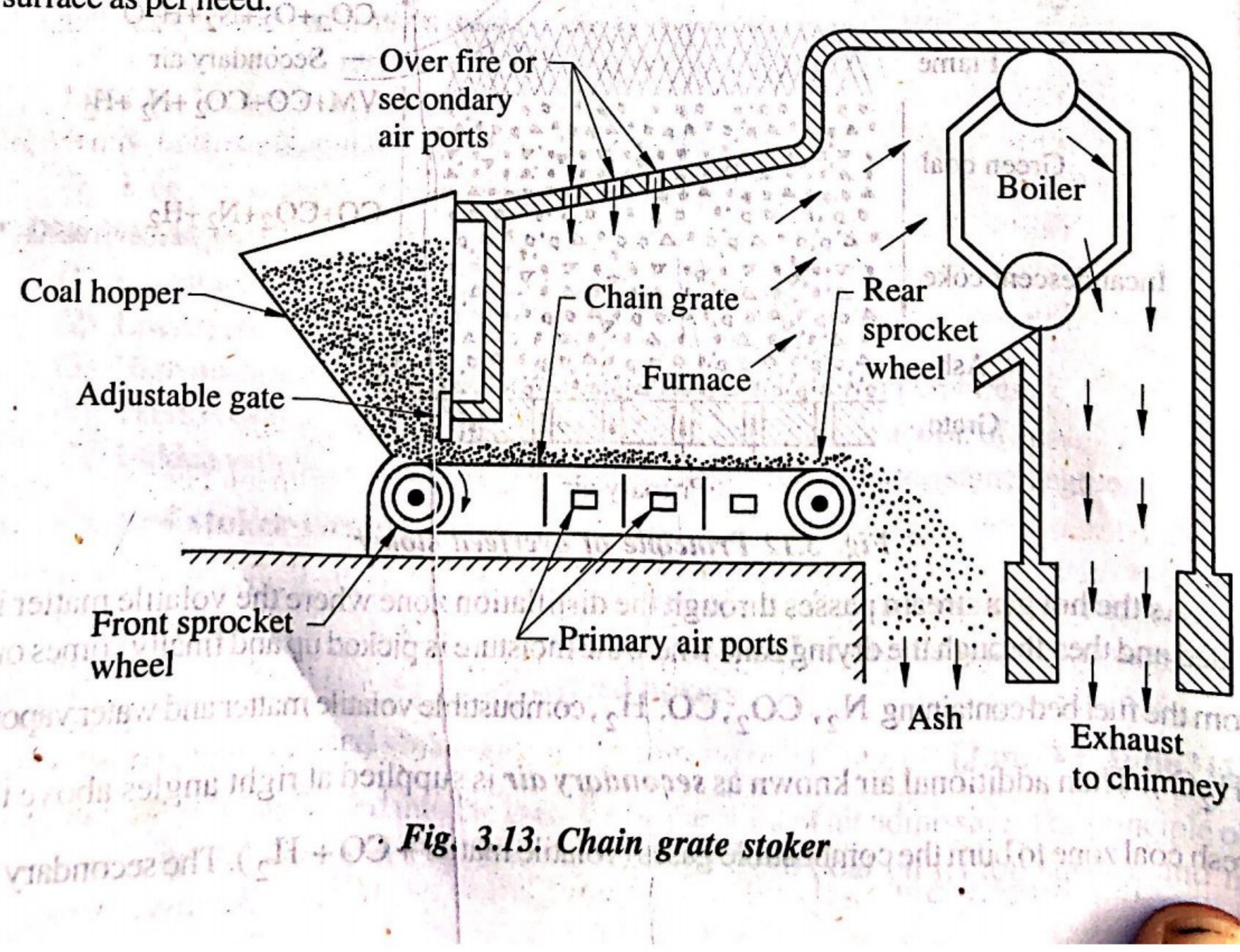


Fig. 3.13. Chain grate stoker

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A travelling type chain grate stoker is shown in Fig. 3.13. The chain grate stoker consists of an endless chain which forms a support for the fuel bed. The chain travels over two sprocket wheels one at the front and other at the rear of furnace. The front sprocket is connected to a variable speed drive mechanism.

The travelling chain receives coal by gravity at its front end through a hopper and carries it into the furnace. The depth of the coal on the grate is regulated by adjustable gate. The speed of the grate varies at the rate at which the coal is fed to the furnace. The ash is tipped from the rear end of chain.

The air required for combustion enters through the air inlets situated below the grate. The secondary air is supplied through the openings provided in the furnace wall above the grate. The combination of primary air and secondary air provide turbulence required for rapid combustion.

These grates are suitable for low grade coal because the fuel must be burnt before it reaches the rear end of the furnace. The rate of burning with this stoker is 200 to 300 kg/m^3 per hour when forced draught is used.

- **Advantages :**
  (1) It is simple in construction.
  (2) Its maintenance cost is low.
  (3) Its initial cost is low.
  (4) It is self-cleaning stoker.
  (5) The heat release rates can be controlled just by controlling the speed of chain.
- **Disadvantages :**
  (1) There is always some loss of coal in the form of fine particles carried with the ashes.
  (2) The temperature of preheated air is limited to 180°C.
  (3) Not suitable for high capacity boilers (200 tonnes/h or more) because the amount of coal carried on the grate is small.
  (4) The clinker troubles are very common.

(2) Spreader stoker :

A spreader stoker is shown in Fig. 3.14. In this stoker, the coal from the hopper is fed on to a feeder which measures the coal in accordance to the requirement. Feeder is a rotating drum fitted with blades. Feeder can be reciprocating rams, endless belts, spiral worms etc. From the feeder the coal drops on to spreader distributor which spread the coal over the furnace. The spreader system projects the coal particles in a continuous stream on to the grate holding an ignited fuel bed. The speed of the feeder is a function of load on the boiler. The fine particles of coal burn in suspension where corse particles burn on the grate. The secondary air is admitted above the fuel bad to promote the turbulence and to complete combustion. The grate may be stationary or travelling as per the capacity of the boiler. High capacity boilers need travelling grate. The grate made of CI bars is connected to a lever through links underneath the grate. In order to allow the ash to fall below the ash pit, the lever is moved back and forth

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which makes the bar rock about the pivot. A variety of coal is burnt over this grate. Spreader stoker is applied to a wide range of boiler sizes 155 MW to 265 MW.

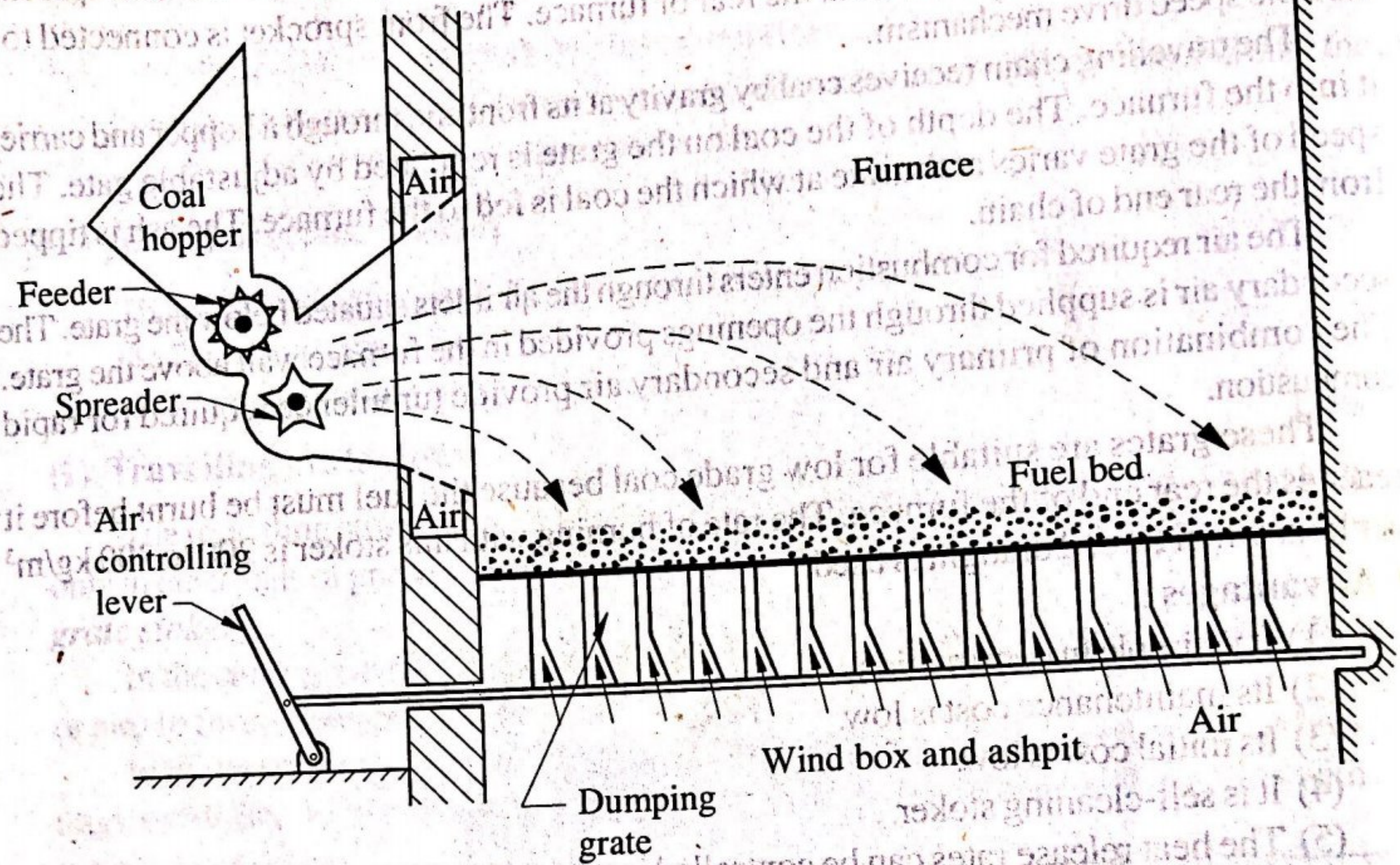


Fig. 3.14 Spreader stoker

- **Advantages :**
  (1) A wide variety of coal can be burnt easily by this stoker.
  (2) A thin fuel bed on the grate is helpful in meeting the fluctuating loads.
  (3) Its operation cost is low.
  (4) The use of high temperature preheated air is possible.
  (5) The clinkering difficulties are reduced even with coals which have high clinkering tendencies by the nature of the spreading action.
- **Disadvantages :**
  (1) The spreader does not work satisfactorily with varying size of coal.
  (2) There is a possibility of some fuel loss because of many fine unburnt carbon particles are also carried with the exhaust gases.
  (3) Fly-ash is much more due to suspension burning of fine fuel particles.

3.7.2 Principle of underfeed stokers [June '11]

Fig. 3.15 shows the underfeed fuel bed showing the various zones. In this type of stokers, the fuel and air move in the same direction. The coal is fed from below the grate by a screw-conveyor or ram. The primary air passing through the holes in the tuyers goes through the various zones. The air mixes with the formed volatile matter and passes through the ignition

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zone and then enters into the region of incandescent coke. When the gas stream passes next through the incandescent coke region, the volatile matter breaks up and readily burns with secondary air fed at the top. The under feed method of fuel supply is best for semi-bituminous and bituminous coals, high in volatile matter, the volatile matter gets heated to a high temperature as it passes through incandescent region of coal. The volatile matter being at a higher temperature before entering the furnace burns quickly when mixed with secondary air.

In case of over feed burning, the volatile matter will be somewhat cooler then the furnace gases and need longer time to ignite and burn.

- **Types of underfeed stoker**

There are mainly two types of underfeed stoker :

(1) Single retort stoker, (2) Multi-retort stoker.

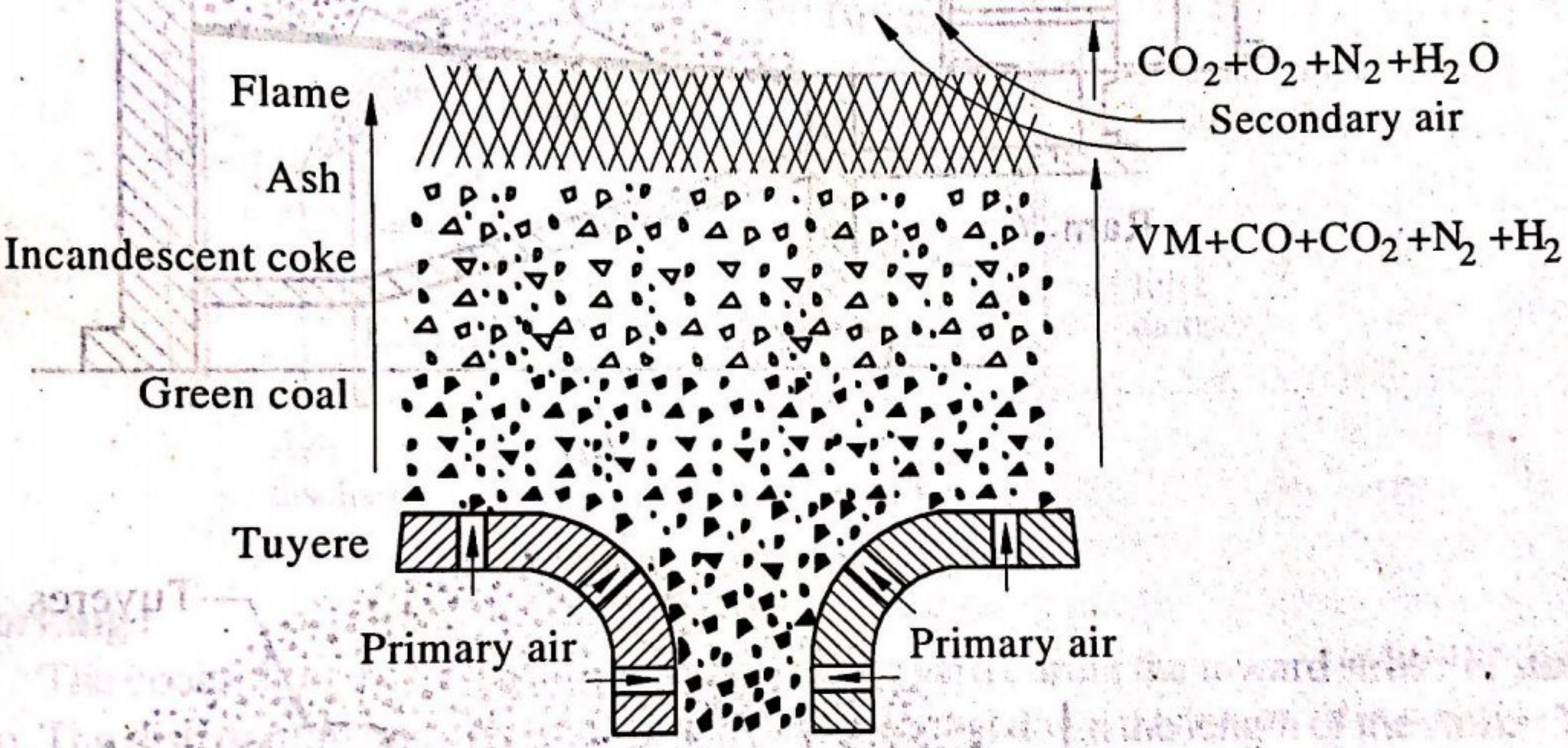
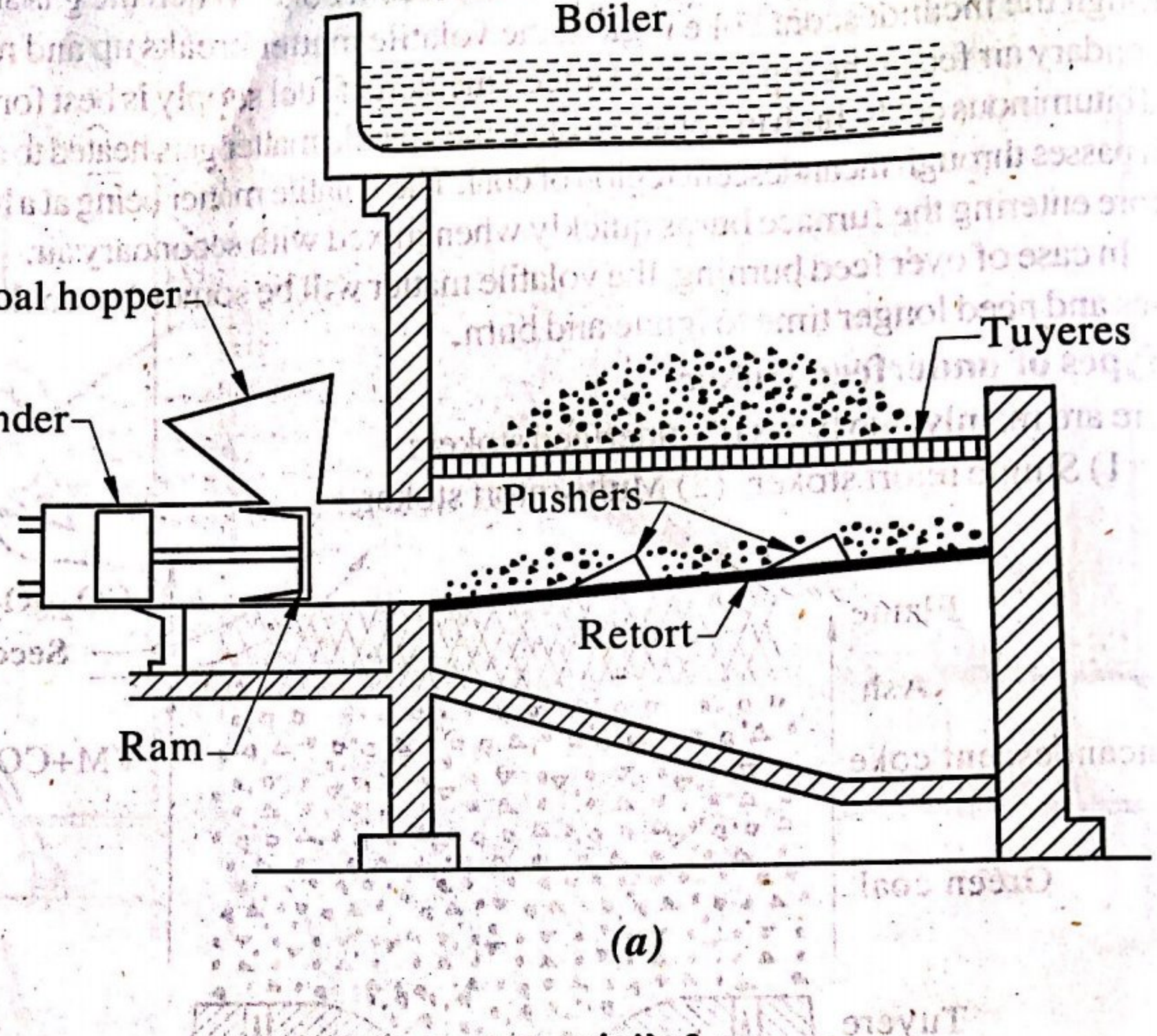


Fig. 3.15 Principle of underfeed stoker.

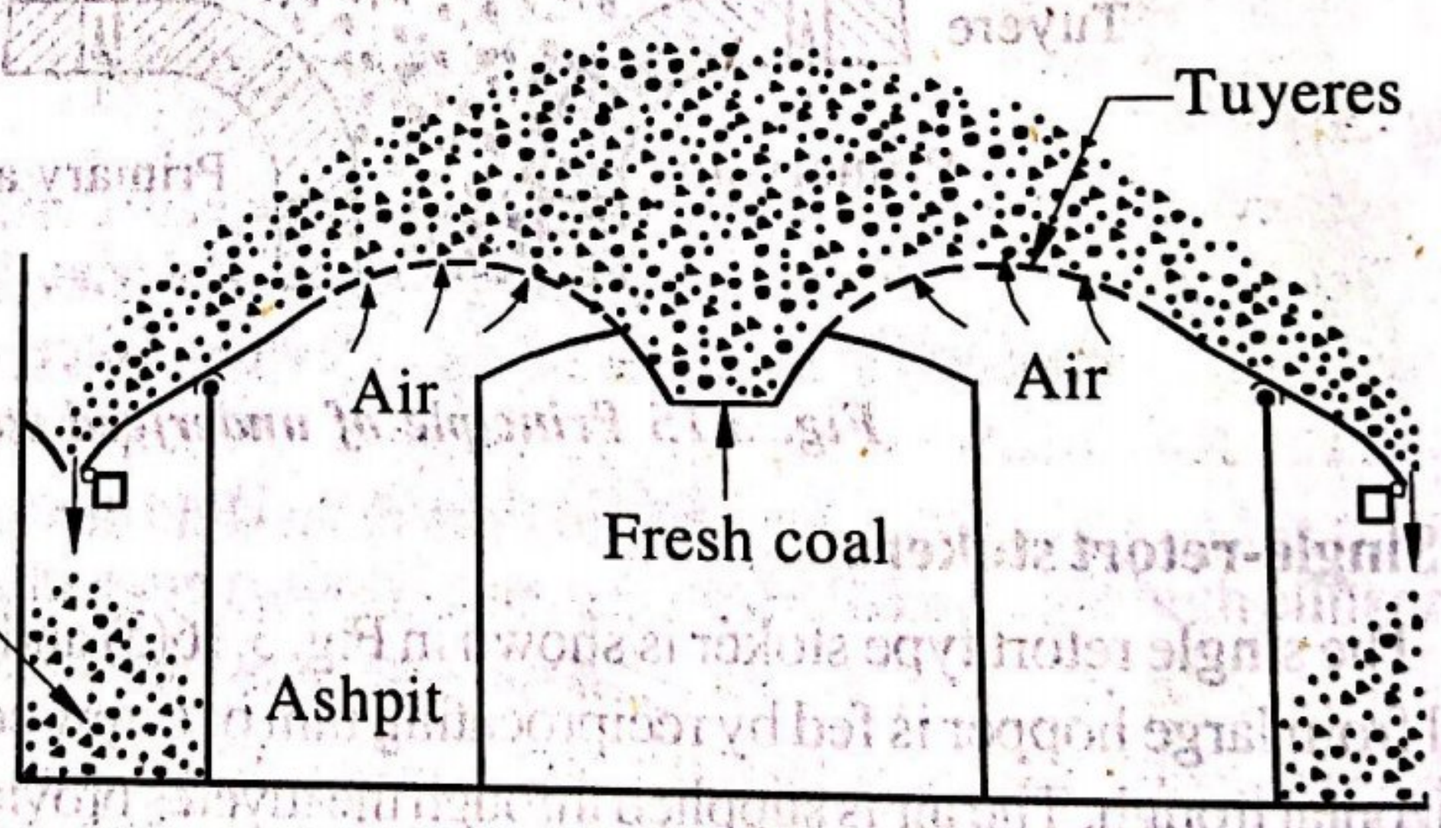
(1) Single-retort stoker :

The single retort type stoker is shown in Fig. 3.16(a) and (b) in form of two views. The coal from large hopper is fed by reciprocating ram or screw conveyor into the bottom of the horizontal trough. The air is supplied through the tuyeres provided along the upper edge of the grate as shown in Fig. 3.16(b). The ash from the grate falls and it is collected in the ash pit. Some of the unburned coal may also pass through the grate. The coal feeding capacity of a single retort stoker varies from 100 to 2000 kg per hour. The size of retort for increasing the capacity is limited by virtue of inability of obtaining even air distribution from the sides of retorts.

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(a)



(b)

Fig. 3.16 Single retort stoker

(2) Multi-retort stoker

A Multi-retort stokers are generally used for increasing the burning capacity of the stoker. Fig. 3.17 shows the multi-retort stoker.

Construction :

It consists of a series of alternate retorts and tuyere boxes for supply of air. Each retort is fitted with a reciprocating ram for feeding and pusher plates for the uniform distribution of

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coal. The number of retorts may vary from 2 to 20 with coal burning capacity ranging from 300 kg to 2000 kg per hour per retort.

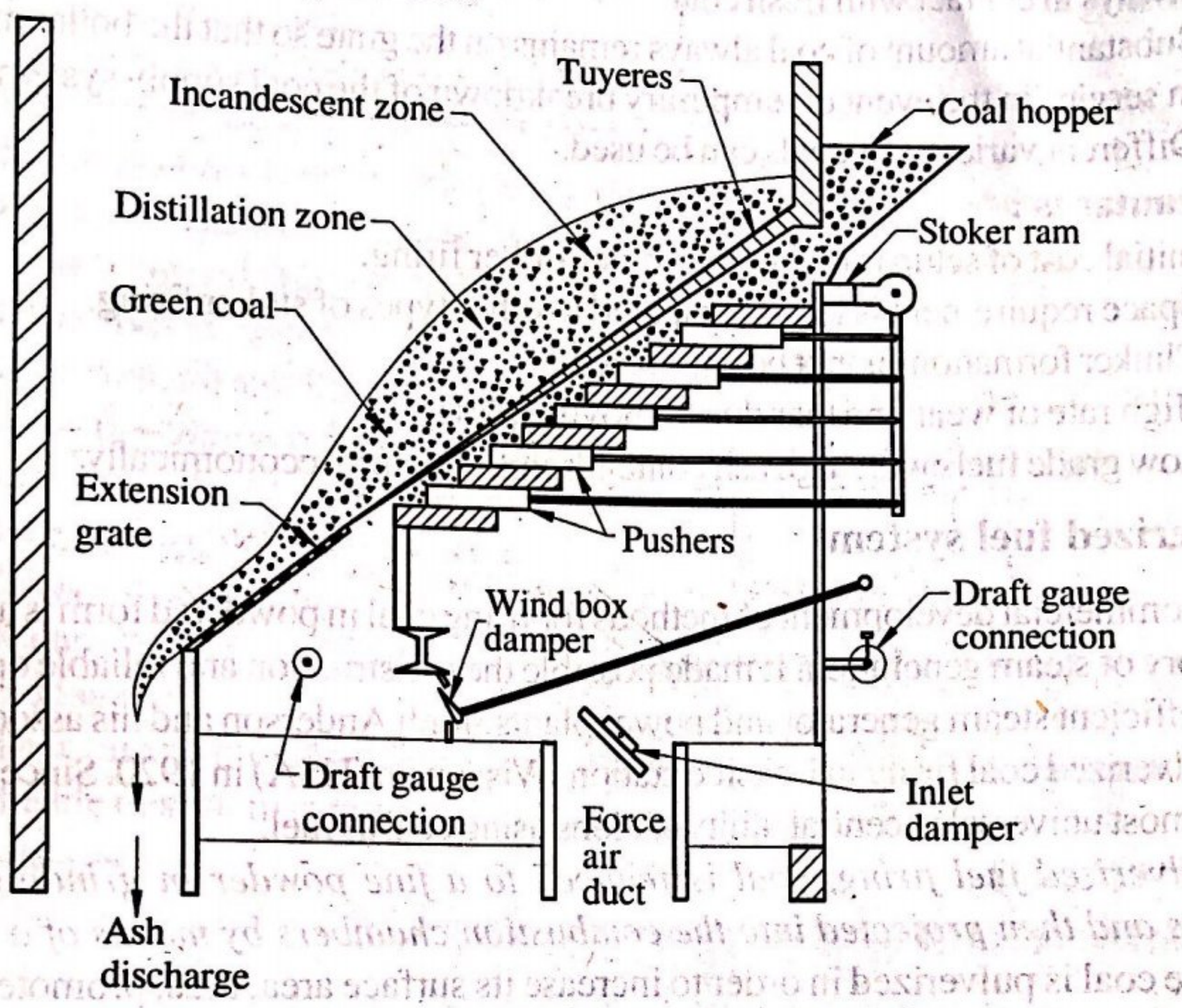


Fig. 3.17 Multi-retort stoker

Working :

The coal falling from the hopper is pushed forward during the inward stroke of stoker ram. The distributing rams (pushers) push the entire coal down the length of the stoker. The stroke length of pushers can be varied as desired. The slope of stroke helps in moving the fuel bed and this fuel bed movement keeps it slightly agitated to break up clinker formation. The primary air is supplied to the fuel bed from main wind box situated below the stoker. The partly burnt coal moves on to the extension grate. The low pressure air entering into the extension grate, wind from main wind box is supplied to the thinner fuel bed on the extension grate. The quantity of air supplied is regulated by an air damper.

These stokers give high combustion efficiency. These stokers can burn large variety of coal efficiently except coals with high ash content.

- **Advantages :**
  - (1) It gives higher combustion efficiency compared with chain grate stokers.
  - (2) The part load efficiency is high.
  - (3) The grate is self-cleaning.
  - (4) Smokeless operation is possible even at very light load.
  - (5) Combustion rate is considerably higher.

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(6) Grate bars, tuyeres and retorts are not subjected to high temperature as they remain always in contact with fresh coal.

(7) Substantial amount of coal always remains on the grate so that the boiler may remain in service in the event of temporary breakdown of the coal supply system.

(8) Different varieties of coals can be used.

- **Disadvantages :**

(1) Initial cost of setup is higher than other stoker firing.

(2) Space requirement is large compared to other types of stoker firing.

(3) Clinker formation cannot be eliminated.

(4) High rate of wear and tear due to moving parts.

(5) Low grade fuels with high ash content cannot be burnt economically.

3.8 Pulverized fuel system

The commercial development of methods for firing coal in powdered form is a land mark in the history of steam generation. It made possible the construction and reliable operation of large and efficient steam generator and power plants. Jonh Anderson and his associates used first the pulverized coal firing in Lakside station, Wisconsin (USA) in 1920. Since then it has become almost universal in central utility stations using coal as fuel.

In pulverized fuel firing, coal is reduced to a fine powder in grinding mills or pulverizers and then projected into the combustion chambers by means of a current of hot air. The coal is pulverized in order to increase its surface area, thus, promotes the rapid combustion without the need of supplying much of excess air, consequently, it results into high thermal efficiency. *The amount of air which is used to dry the coal and convey the powdered fuel to the furnace is known as* ***primary air*** and *the amount of air which is blown in separately to complete the combustion is known as* ***secondary air.***

To burn pulverized coal successfully in a furnace, two requirements must be met :

(i) the existence of large quantities of very fine particles of coal, usually those that would pass a 200-mesh (0.074 mm) screen, to ensure ready ignition because of their large surface to volume ratios, and

(ii) the existence of a minimum quantity of coarser particles to ensure high combustion efficiency.

The process of coal pulverization is carried out in the two stages :

Stage-I : Raw and lump coal is crushed to a particle size not more than 15-25 mm in the crusher.

Stage-II: Crushed coal is delivered into raw coal bunkers and from here it is transferred to grinding mills that grind the feed into the final particles of 200-300 mesh size. During grinding hot air is blown through the fuel to dry it to impart good fluidity of the coal dust.

- **Advantages of pulverized coal firing :** **[Dec. '13, June '11, Dec. '10]**

(1) Large increase in surface area of coal per unit mass allows faster and efficient

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combustion as more coal surface is exposed to heat and oxygen.
(2) Less excess air is required for complete combustion because of greater surface area of fuel exposed.
(3) Any type of coal right from anthracite to peat can be successfully burnt.
(4) Better response to variation in load due to better combustion control.
(5) Large amount of heat release makes it very suitable for super thermal power stations where the rate of steam generation is as high as 2000 t/h.
(6) Clinkering and slagging problems are low.
(7) Carry over of unburnt fuel to ash is practically nil.
(8) It has low ash handling problem.
(9) It can operate successfully in combination with gas and oil fired system.
(10) Cold start-up of boilers is very rapid and efficient.
(11) Less furnace volume is required.
(12) It has smokeless operation.
(13) It can use highly preheated air for combustion which helps for rapid flame propagation.
(14) There are no moving parts in the furnace subjected to high temperature, therefore the life of system is more and the operation is troubleless.

- **Disadvantages :** **[Dec. '13, Dec. '10]**

(1) The capital cost is high as it requires many additional equipments compared to stoker firing.
(2) Its operation cost is high compared to stoker firing.
(3) Lot of fly ash in the exhaust which makes the removing of fine dust uneconomical.
(4) Since coal will burn like a gas, there will always be danger of explosions.
(5) Special equipment is required for starting.
(6) The maintenance of furnace brick work is costly because of higher working temperature.
(7) The skilled operators are required.
(8) High air pollution due to the emission of very fine particles of grit and dirt.

- **Comparision between pulverized coal firing and stoker firing :**

Pulverized coal firing	Stoker firing
1. Coal is to be supplied in powder form to furnace.	1. Coal is to be supplied in solid form to furnace.
2. Powder form of coal burns like a gas	2. Coal burns on grate.
3. It can be used for large capacity (more than 100 MW) power plant.	3. It can be used for low to medium capacity power plant.
4. Less amount of excess air is required for complete combustion	4. More excess air is required for complete combustion.
5. More building space is required.	5. Less building space is required.

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6. The maintenance and operating costs are more.	6. The maintenance and operating costs are less.
7. The possibilities of explosion are more.	7. It is free from danger of explosion.
8. More efficient combustion	8. Less efficient combustion.

3.9 Pulverized fuel handling systems

Pulverized fuel handling system is a family of equipment in which coal is ground, dried and fed to the burners of a boiler furnace.

In general, there are two methods used to feed the pulverized fuel to the combustion chamber of the boiler as follows :

1. Unit system, 2. Central or bin system

3.9.1 Unit system

In this system, each boiler is provided with its own pulverizing unit while certain provisions are made to transfer the pulverized coal to neighbouring boilers to increase the reliability of the fuel supply.

The arrangement of unit system is shown in Fig. 3.18. Crushed coal from raw coal bunker is fed to the pulverizer through feeder at a variable rate governed by the combustion requirements of boiler furnace and steam generation rate. Hot air or flue gases are passed through the feeder to dry the coal before feeding to the pulverizer. The pulverized coal is carried from the mill with the help of Induced Draft (I.D.) fan via a separator and blown into the furnace along with the hot air. In the separator, the big particles of coal are separated from the fine dust and these again fall down into the mill. The secondary air is supplied to the burner before entering the fuel into the combustion chamber.

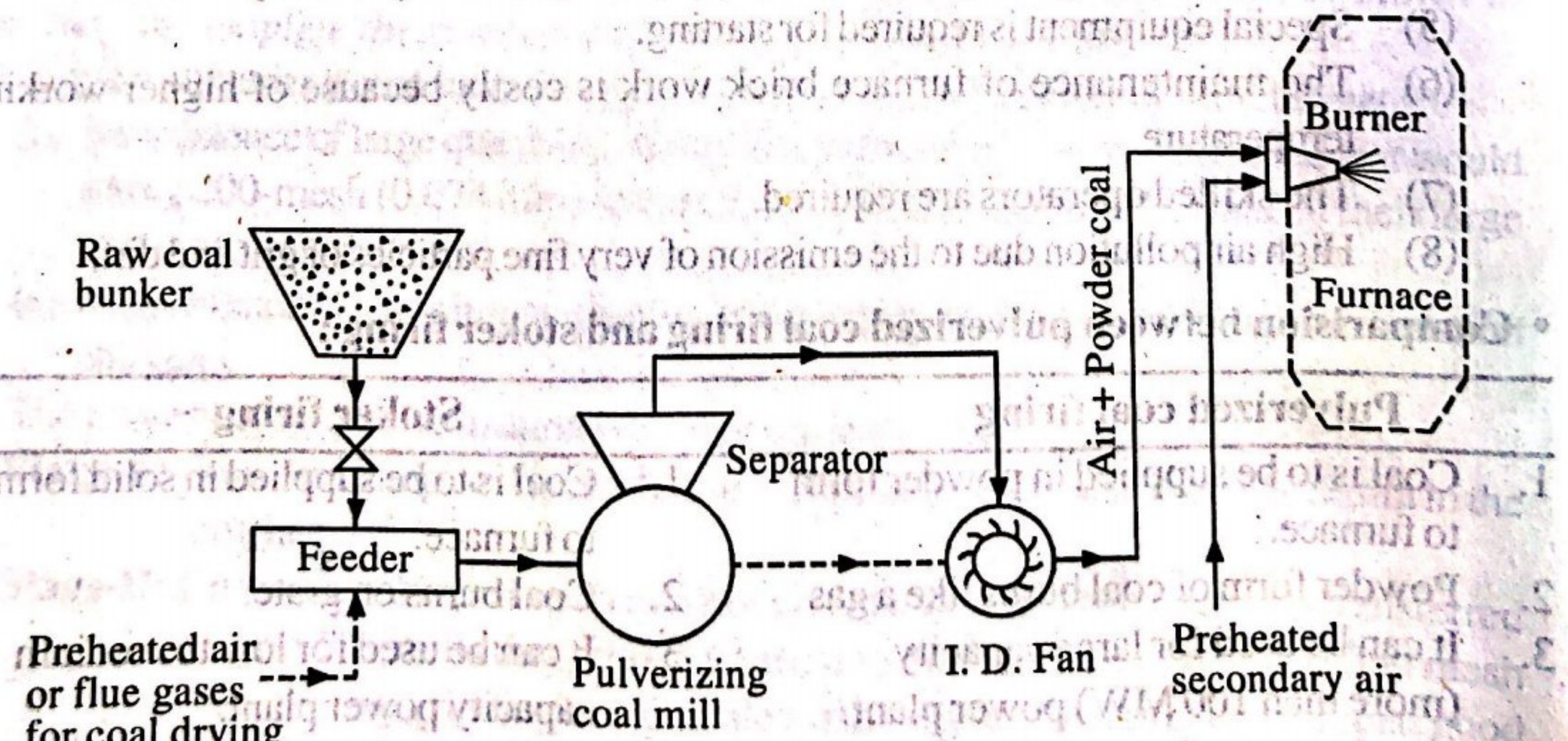


Fig. 3.18 Unit system

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Advantages :

(1) Simpler in lay-out, design and operation.
(2) Initial cost is less compared to central system.
(3) Allows direct combustion control from the pulverizer.
(4) Lower maintenance charge.
(5) Less space is required.
(6) No drying unit is required.

Disadvantages :

(1) If one pulverizing unit goes out of order, its corresponding boiler unit has to be shutdown.
(2) The pulverizing unit operates at variable load as per the load on the power plant which results in poor performance at part load.
(3) Lesser degree of flexibility.
(4) There is greater wear off an I.D. fan blades as these handle both air and abrasive coal particles.

3.9.2 Central (Bin) System

In this system coal is pulverized on a centralized basis, the pulverized fuel is stored in a central bin wherefrom it is distributed through pipelines between the boilers.

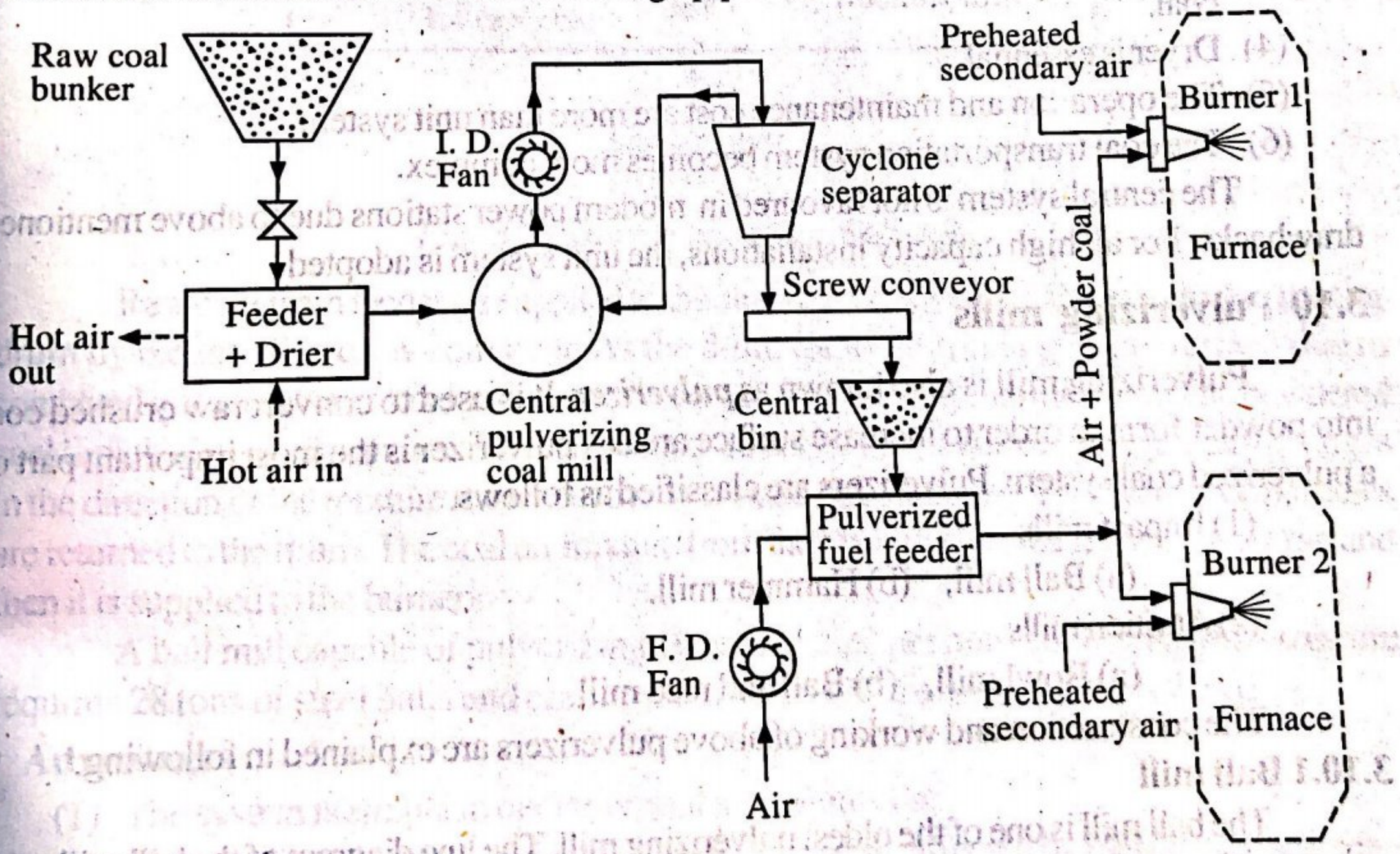


Fig. 3.19 Central or Bin System

The arrangement of central feed system is shown in Fig. 3.19. The coal from raw coal bunker is fed to the drier. The coal drying is achieved by using hot gases, preheated air or bled

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steam. Feeder is used to feed dried coal to the pulverizer. The pulverized coal is carried from the pulverizer mill with the help of air and it is separated in the cyclone separator. The separated pulverized coal is transferred to the central bin (bunker) with the help of conveyor. Forced Draft (F.D.) fan is used to supply primary air to feeder. The mixture of pulverized coal and air is then supplied to the burners as shown in Fig. 3.19.

- **Advantages :**
  (1) Greater flexibility and better response to abrupt load variation.
  (2) Less power consumption per ton of coal pulverized.
  (3) Operation of burners is independent of coal preparation.
  (4) The fan handles only air therefore there is no problem of excessive wear of fan blades.
  (5) Pulverizer can be shutdown when there is enough reserve of pulverized coal.
  (6) It offers good control over the fineness of coal.
  (7) It requires less labour.
  (8) The pulverizer capacity is low as it always works at rated load.
- **Disadvantages :**
  (1) Initial cost of central system is high.
  (2) It occupies more floor space.
  (3) The possibility of fire hazard is more due to storage of large amount of pulverized coal.
  (4) Dryer is essential.
  (5) The operation and maintenance cost are more than unit system.
  (6) The coal transportation system becomes more complex.

  The central system is not favoured in modern power stations due to above mentioned drawbacks. For all high capacity installations, the unit system is adopted.

3.10 Pulverizing mills

Pulverizing mill is also known as ***pulverizer***. It is used to convert raw crushed coal into powder form in order to increase surface area. A pulverizer is the most important part of a pulverized coal system. Pulverizers are classified as follows.

(1) Impact mills
  (a) Ball mill, (b) Hammer mill.
(2) Attrition mills
  (a) Bowl mill, (b) Ball and race mill.

The construction and working of above pulverizers are explained in following.

3.10.1 Ball mill

The ball mill is one of the oldest pulverizing mill. The line diagram of the ball mill using two classifiers is shown in Fig. 3.20. It basically consists of a hollow cylinder with conical ends and heavy-cast wear-resistant liners. The cylinder is filled less than half with forged steel balls of mixed size.

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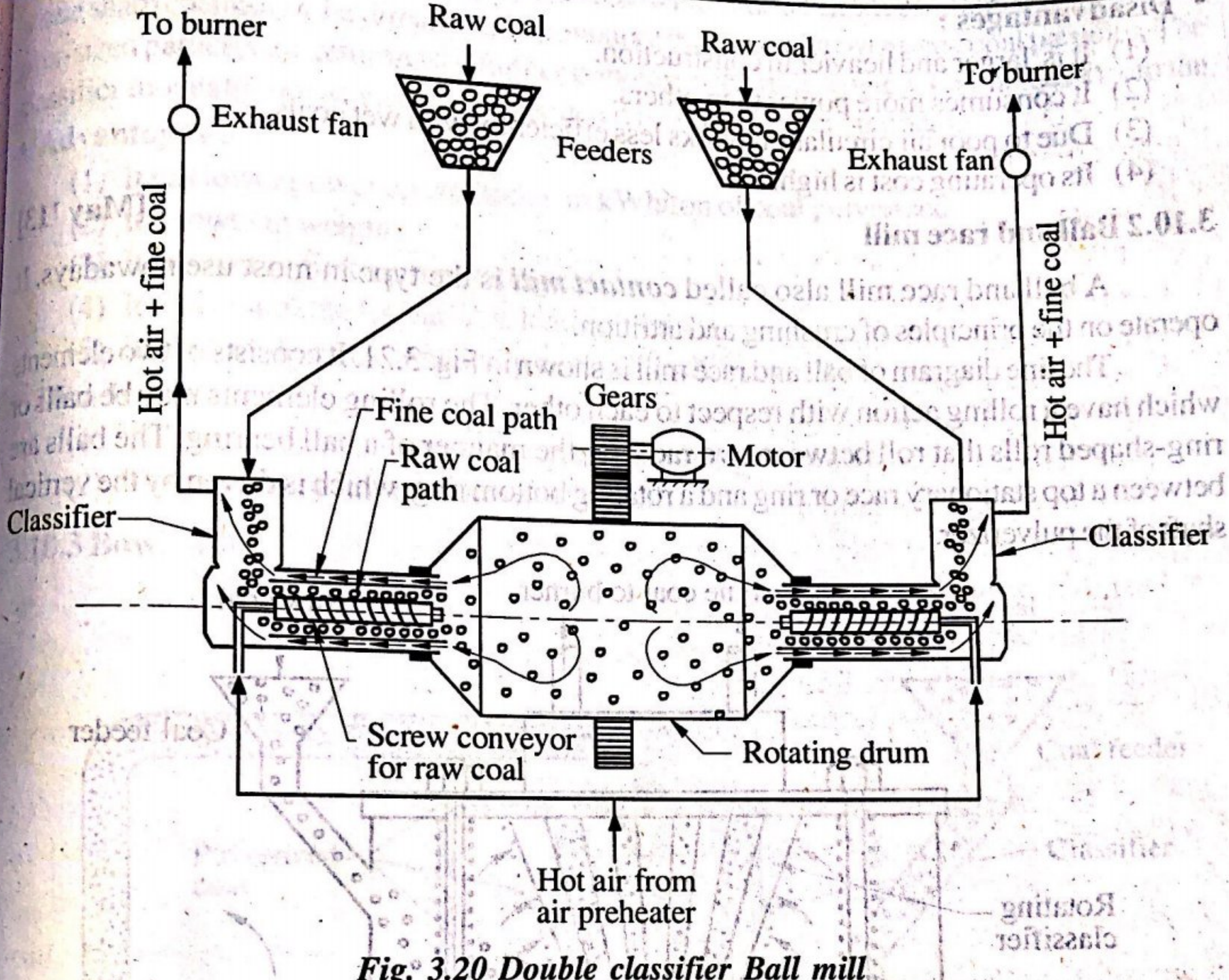


Fig. 3.20 Double classifier Ball mill

Raw coal from feeders is supplied to the classifiers from where it moves to the rotating drum by means of a screw conveyor. As the drum rotates the coal gets pulverized due to combined impact between coal and steel balls. Hot air is introduced into the drum. The powdered coal is picked up by the air and the coal-air mixture enters the classifiers, where sharp changes in the direction of the mixture throw out the over sized coal particles. The over-sized particles are returned to the drum. The coal air mixture from the classifier moves to the exhaust fan and then it is supplied to the burners.

A ball mill capable of pulverizing 10 tons of coal per hour containing 4% moisture requires 28 tons of steel balls and consumes 20 to 25 kW h energy per ton of coal.

- **Advantages :**
  - (1) The system is simple in operation and low initial cost.
  - (2) The grinding elements in this mill are not seriously affected by metal scrap and other foreign material in the coal.
  - (3) Suitable for a wide range of fuels including anthracite and bituminous coal.
  - (4) Low maintenance cost.

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- **Disadvantages :**
  (1) It is larger and heavier in construction.
  (2) It consumes more power than others.
  (3) Due to poor air circulation works less efficiently with wet coals.
  (4) Its operating cost is high.

[May '13]

3.10.2 Ball and race mill

A ball and race mill also called ***contact mill*** is the type in most use nowadays. It operate on the principles of crushing and attrition.

The line diagram of ball and race mill is shown in Fig. 3.21. It consists of two elements which have a rolling action with respect to each other. The rolling elements may be balls or ring-shaped rolls that roll between two races, in the manner of a ball bearing. The balls are between a top stationary race or ring and a rotating bottom ring, which is driven by the vertical shaft of the pulverizer.

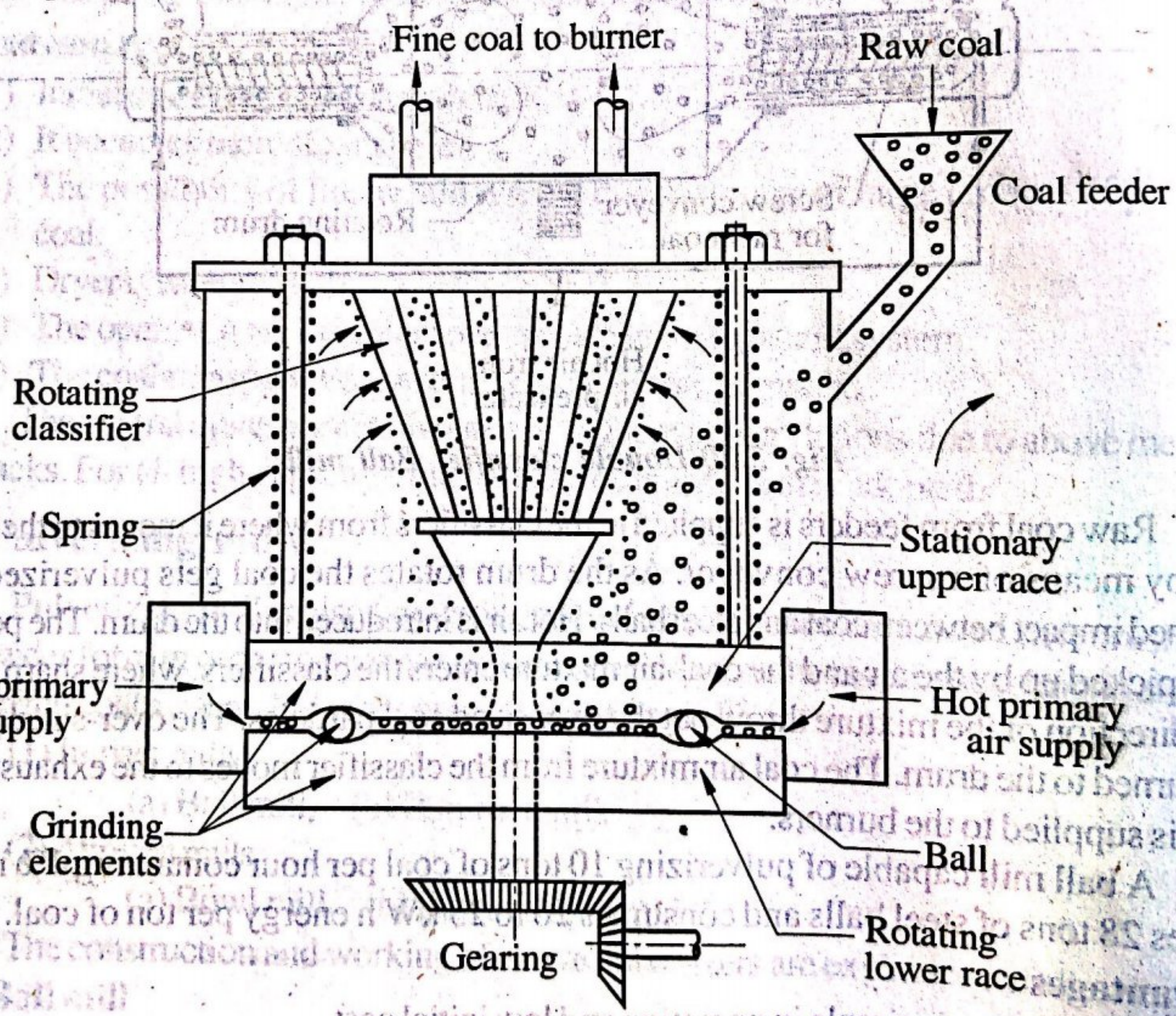


Fig. 3.21 Ball and race mill

The coal is supplied through the rotating table feeder at the upper right to fall on the inner side of the races. The moving balls and races catch coal between them to crush it to a powder. Spring hold down the upper stationary race and adjust the force needed for crushing. Hot air is supplied to the mill through the annular space surrounding the races by a forced draft

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fan. The powdered coal is picked up by the air and the coal-air mixture enters the classifiers, where sharp changes in the direction of the mixture throw out the over sized coal particles. The over-sized particles are returned for further grinding in the mill. The coal-air mixture from the classifier moves to the exhauster fan and then it is supplied to the burners.

- **Advantages :**
  (1) It has lower power consumption in kWh/ton of coal pulverized.
  (2) It is lower in weight.
  (3) It is lower in first and operating cost.
  (4) It is also suitable for variable load conditions.
  (5) It can handle coals containing as much as 20% moisture.
- **Disadvantages :**
  (1) This mill has greater wear compared to other pulverizers.
  (2) The leakage of fine coal through the mill.

3.10.3 Bowl mill

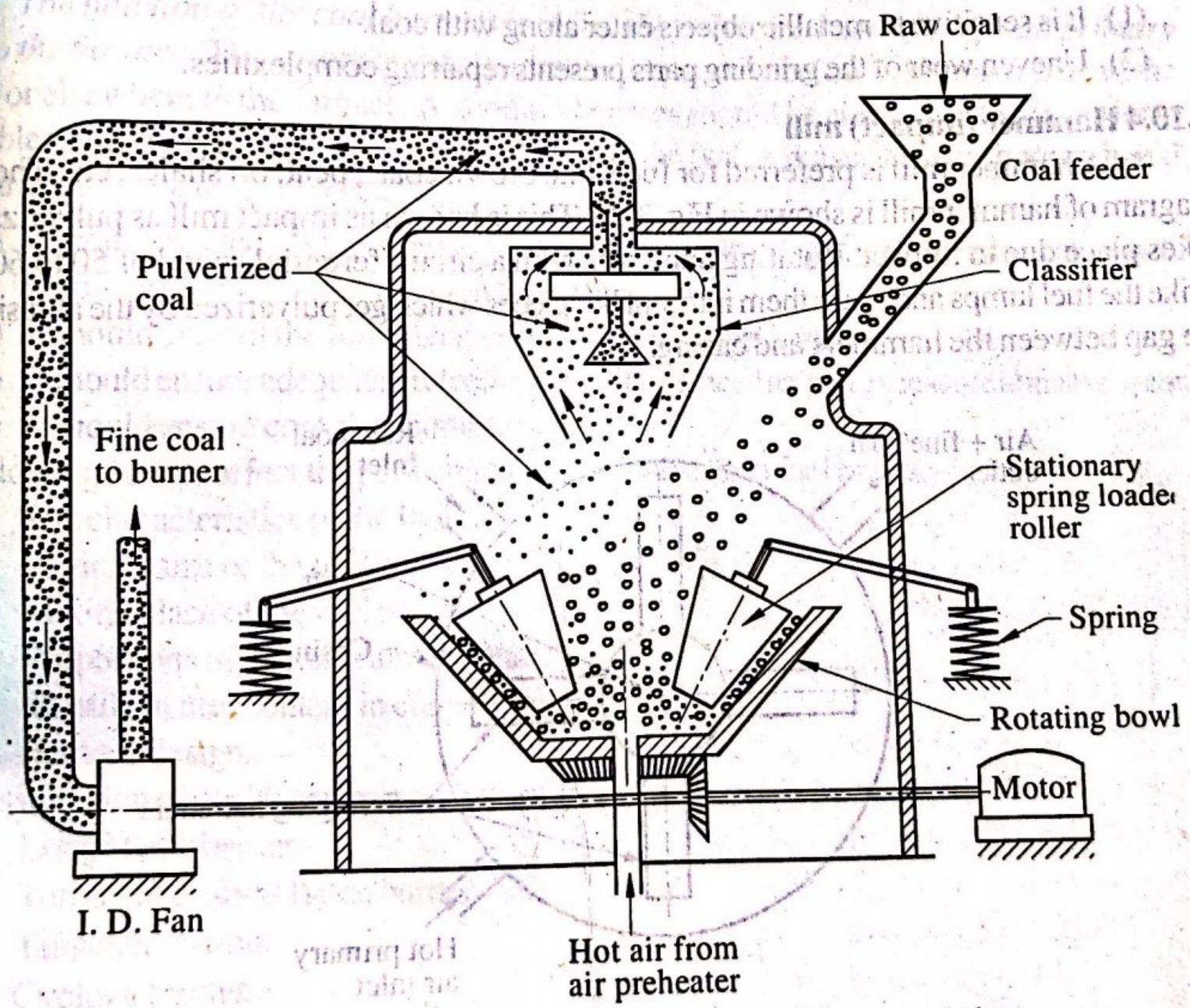


Fig. 3.22 Bowl mill

The bowl mill is widely used for pulverizing the coal. The line diagram of bowl mill is shown in Fig. 3.22. It consists of stationary rollers mounted on an electrically driven rotating

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bowl. Coal fed through the hopper gets pulverized by attrition as it passes between the sides of the rollers and bowl.

Hot primary air introduced into the pulverizer through the bottom of the bowl carries off coal dust into the centrally located classifier fitted at the top. Coarse particles drop back into the bowl through the centre cone of the classifier while the fine coal dust-air mixture is led away to the burner.

- **Advantages :**
  (1) It has lower energy consumption (12-15 kWh/t).
  (2) Less over all dimensions.
  (3) The classifier can be adjusted to alter the degree of coal fineness while the mill is on.
  (4) Leakage of coal from the mill casing is practically nil because it operates at negative pressure.
  (5) It produces less noise.

- **Disadvantages :**
  (1) It is sensitive to metallic objects enter along with coal.
  (2) Uneven wear of the grinding parts presents repairing complexities.

3.10.4 Hammer (impact) mill

A hammer mill is preferred for fuels like brown coals, peat, oil shales, etc. The line diagram of hammer mill is shown in Fig. 3.23. This is known as impact mill as pulverization takes place due to impact. Rotating hammers with a circumferential speed of 50 to 60 m/s strike the fuel lumps and crush them into smaller pieces which get pulverized by the abrasion in the gap between the hammers and casing.

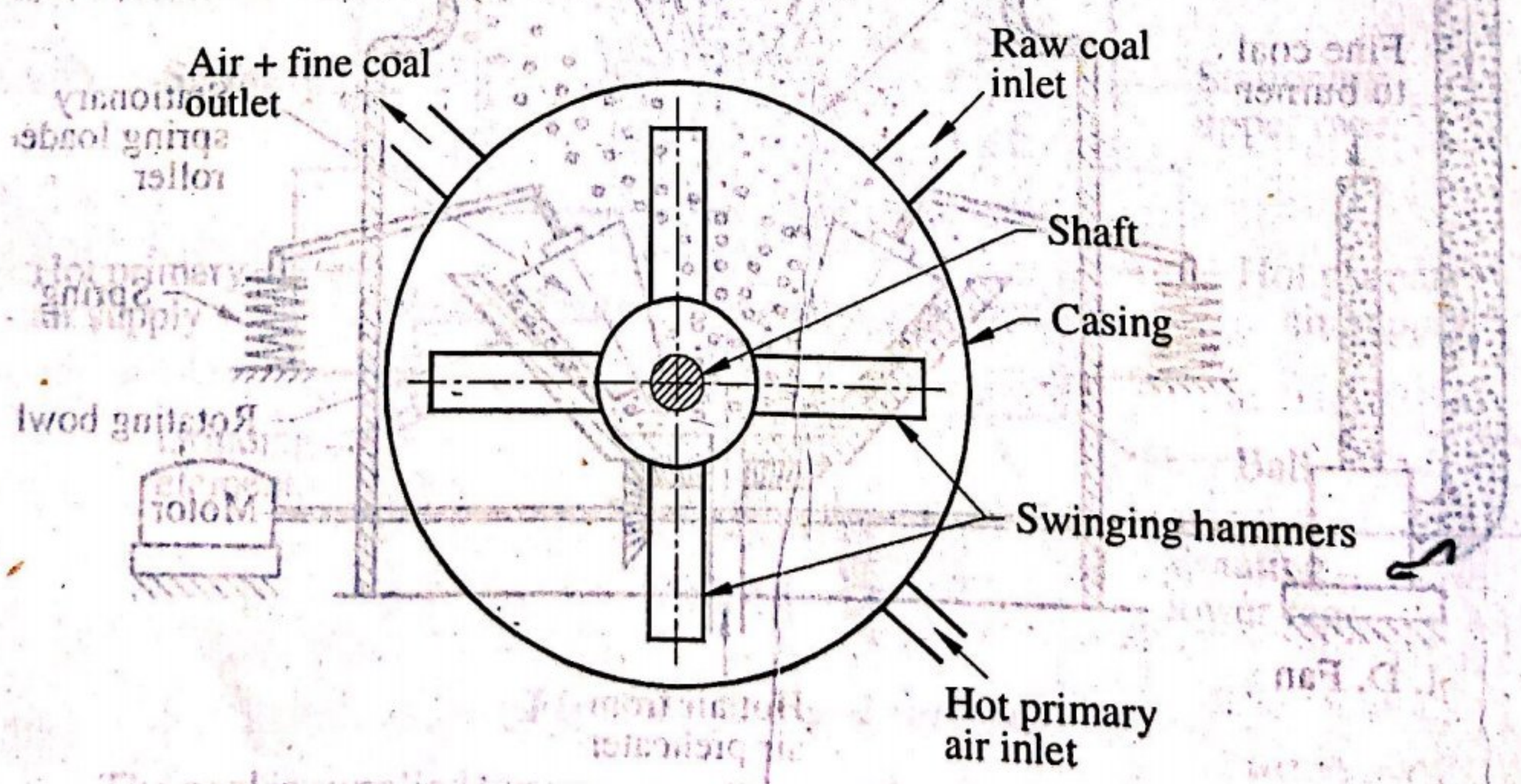


Fig. 3.23 Hammer mill

The primary air fan induces air through the pulverizer, a flow of air that lifts the coal

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dust. The airborne fuel dust is subjected to a centrifugal dust separator to throw the oversize particles back into the grinding section while the finely divided fuel particles suspended in primary air are discharged through a centrally located dust discharging duct.

- **Advantages :**
  (1) It is simple and low in capital cost.
  (2) It requires minimum floor area.
  (3) Its operational speed is high.
  (4) Wet coal can be pulverized.
- **Disadvantages :**
  (1) Power consumption is high when fine powder is required.
  (2) Its capacity is limited.
  (3) Care is required to prevent metal scrap entering to the pulverizer mill.

3.11 Pulverized coal burners

The function of the coal burners is to fire the pulverized coal along with primary air into the furnace. The secondary air is admitted separately below the burner, around the burner or elsewhere in the furnace. A good coal burner should be able to produce a uniform and stable flame with almost complete combustion of the fuel. A pulverized coal burner should satisfy the following requirements :

(1) It should mix the coal and primary air thoroughly and bring this mixture in contact with secondary air to create sufficient turbulence.
(2) It should control the flame shape and its travel in the furnace.
(3) It should ensure adequate protection against overheating and excessive abrasive wear.
(4) It should ensure complete combustion.

The following factors affect the performance of the pulverized fuel burner :

(1) The characteristics of the fuel used.
(2) Particles size of the pulverized coal.
(3) Mixing place of the fuel and air.
(4) Proportions of primary and secondary airs.
(5) Volatile matter content in coal.
(6) Furnace design.

The classification of coal burners is as follows :

(1) Long-flame burner.
(2) Turbulent or short flame burner.
(3) Tangential burner.
(4) Cyclone burner.

3.11.1 Long flame or U-flame or streamlined burner [Jan. '13]

The arrangement of long flame burner is shown in Fig. 3.24. The pulverized coal along

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with primary air is supplied to combustion chamber as shown in Fig. 3.24. ***Tertiary*** air is supplied around the burner to form an envelope around the primary air and fuel to provide better mixing. Hot secondary air is introduced at right angles to the flame which provides necessary mixing for better and rapid combustion. In this burner, the flame is required to travel a considerable distance as shown due to which it allows sufficient time to complete the combustion of fuel.

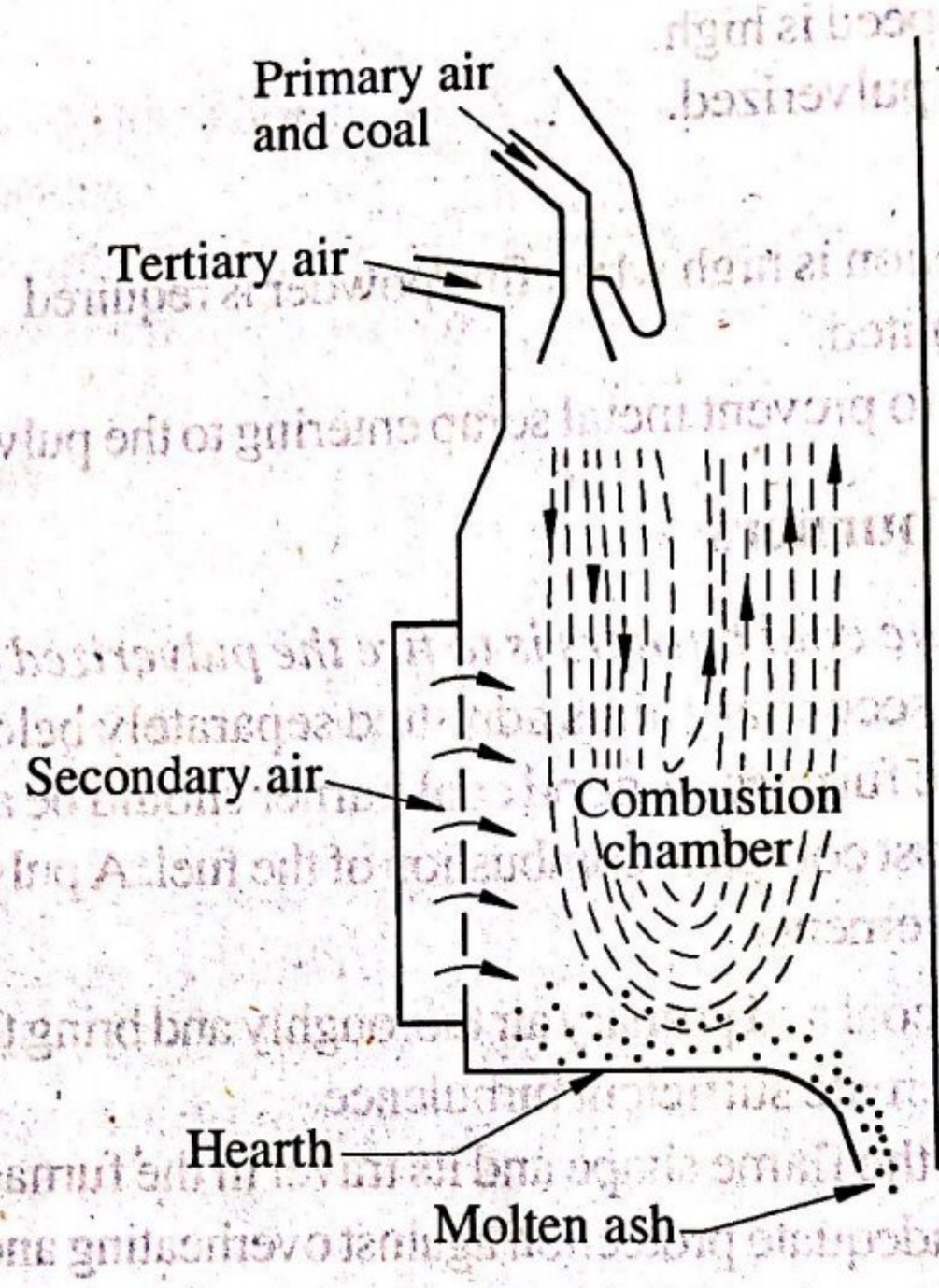


Fig. 3.24 Long flame burner

For low volatile coal, combustion chambers are equipped with such burners to give a long flame path for slower burning of coal particles. The longer path provides more time to burn and it is required to control the velocity in this zone.

3.11.2 Short flame or turbulent burner [Jan. '13]

Turbulent burner can fire horizontally or at some inclinations by adjustment. The pulverized coal-air mixture and secondary hot air are arranged to pass through the burner in such a way that there is good mixing and the mixture is projected in highly turbulent form in the furnace. Due to high turbulence created before entering the furnace, the mixture burns as soon as it enters, and combustion is completed in short distance.

The arrangement of this burner is shown in Fig. 3.25. This burner gives high rate of combustion compared with other type. This is generally preferred for high volatile coals. All modern power plants use this type of burner.

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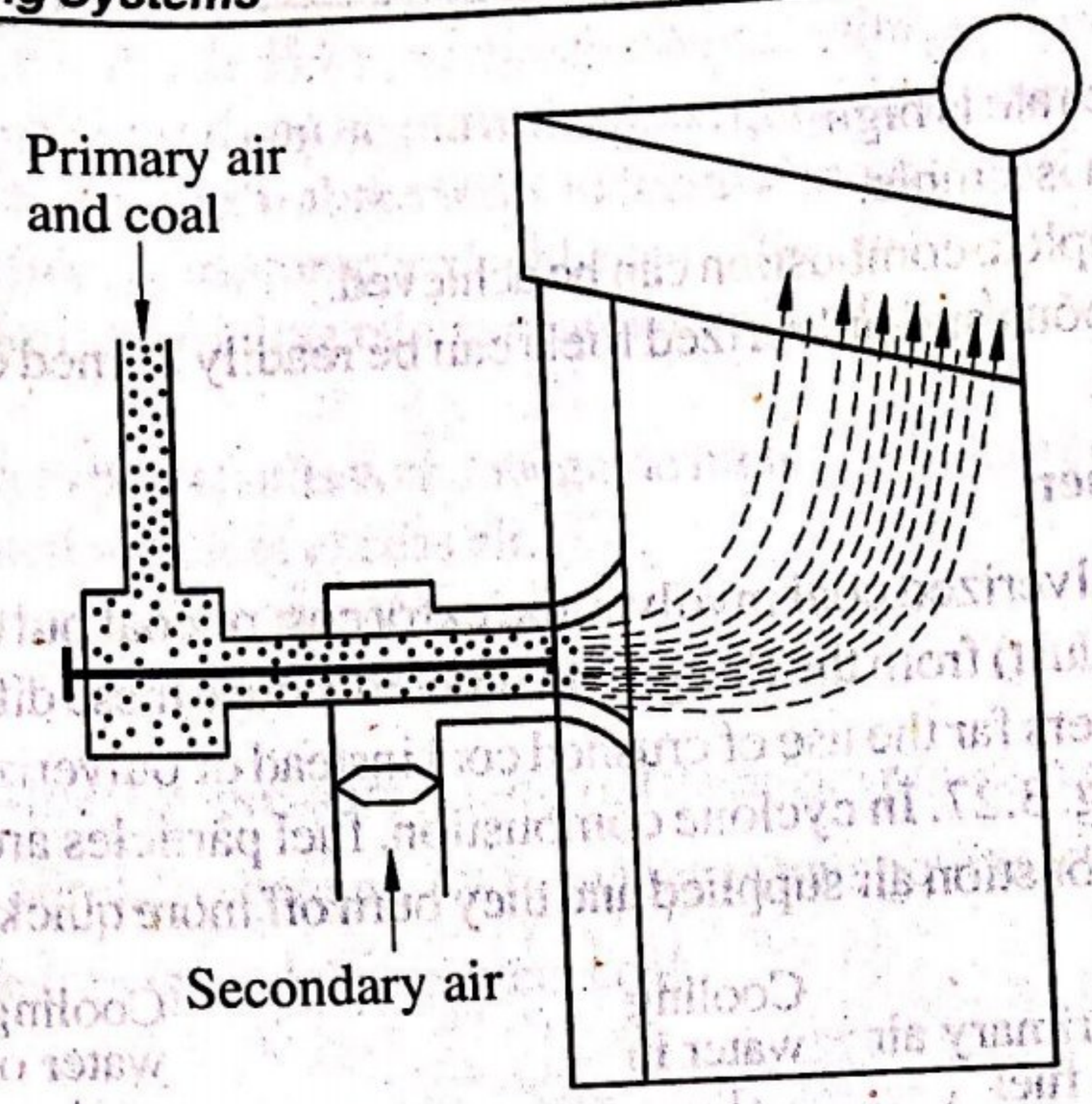


Fig. 3.25 Turbulent burner

[Jan. '13]

3.11.3 Tangential burners

The arrangement of these burners are shown in Fig. 3.26. In this case four burners are located in the four corners of the furnace and are fires in such a way that the four flames are tangential to an imaginary circle formed at the centre. The swirling action produces sufficient turbulence in the furnace to complete the combustion in a short period and avoids the requirement of producing high turbulence at the burner it self. With this method of firing, high heat release rates are possible.

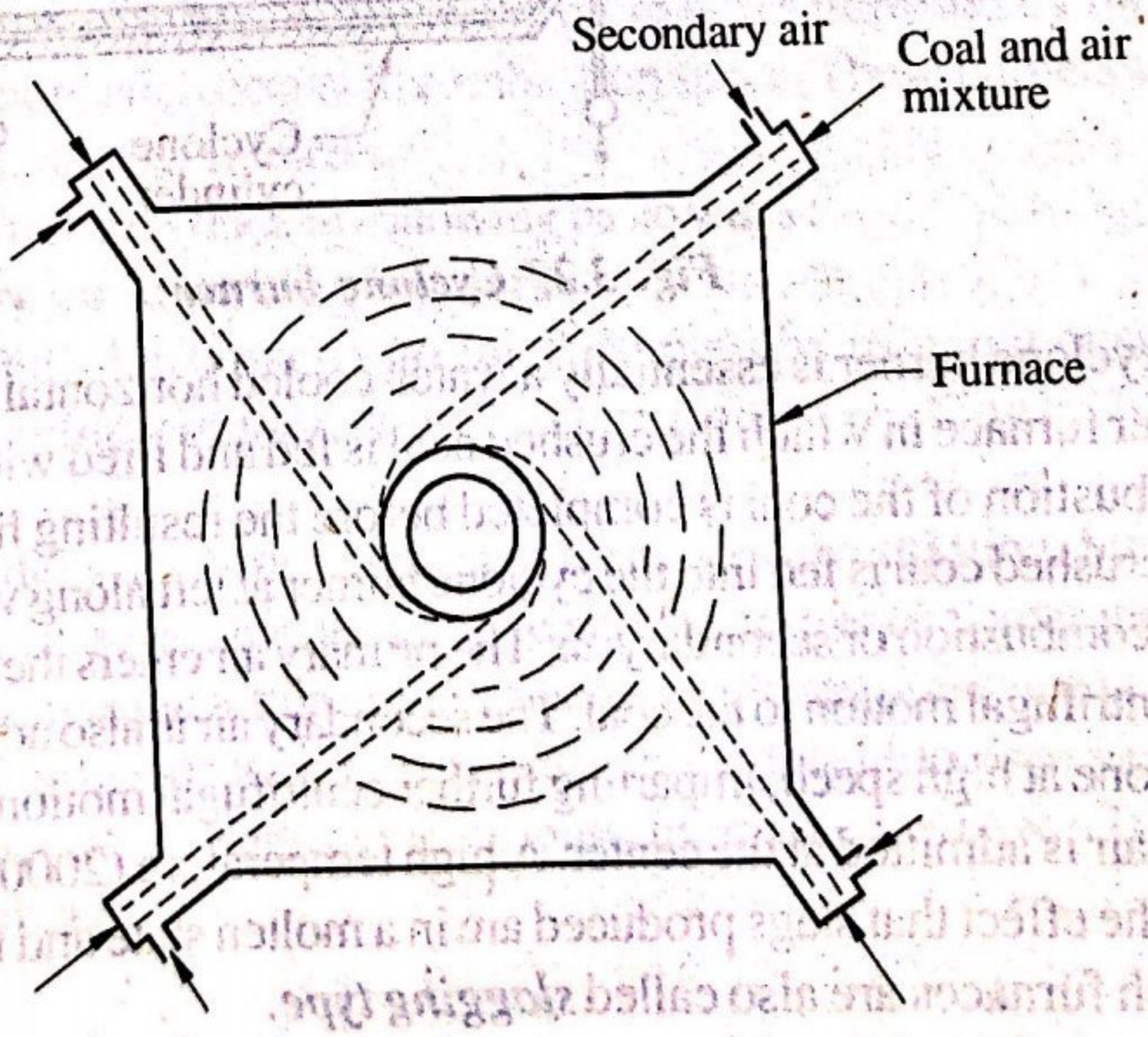


Fig. 3.26 Tangential burners

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- **Advantages :**
  (1) Heat release rate is high.
  (2) Its operation is simple.
  (3) Almost complete combustion can be achieved.
  (4) Liquid, gaseous and pulverized fuels can be readily burned either separately or in combination.

[May '13]

3.11.4 Cyclone burner

The use of pulverized coal involves the extra cost of coal pulverization and cost of collection of fly-ash (dust) from the flue gas. In order to reduce these difficulties an attempt is made to develop burners far the use of crushed coal instead of pulverized coal. The cyclone burner is shown in Fig. 3.27. In cyclone combustion, fuel particles are subjected to a great turbulence by the combustion air supplied and they burn off more quickly.

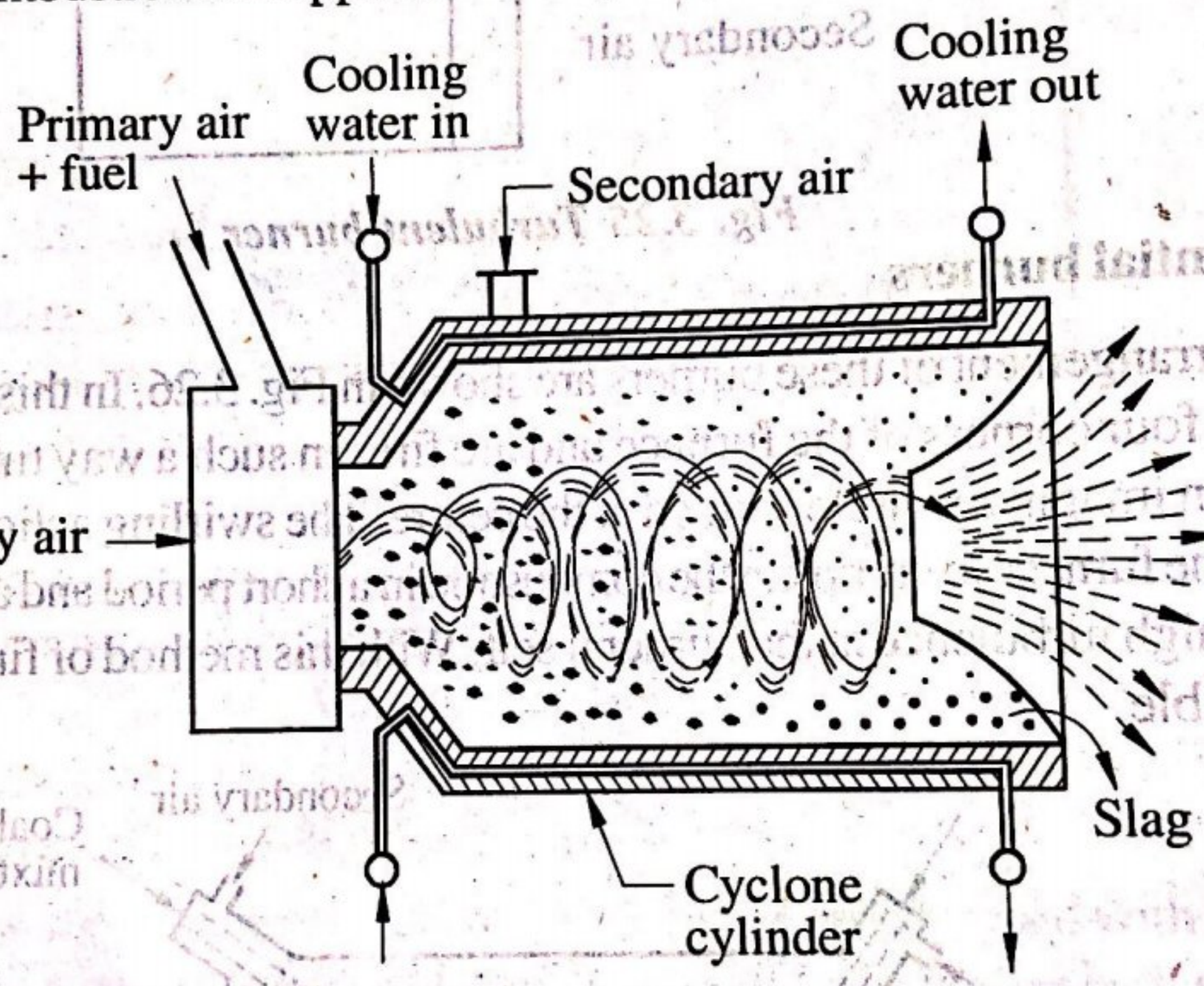


Fig. 3.27 Cyclone burner

The cyclone burner is essentially a water-cooled horizontal cylinder located outside the main boiler furnace in which the crushed coal is fed and fired with very high rates of heat release. Combustion of the coal is completed before the resulting hot gases enter the boiler furnace. The crushed coal is fed into the cyclone burner at left along with primary air, which is about 20% of combustion or secondary air. The primary air enters the burner tangentially, thus imparting a centrifugal motion to the coal. The secondary air is also admitted tangentially at the top of the cyclone at high speed, imparting further centrifugal motion. A small quantity of air, called tertiary air is admitted at the center. A high temperature (2000°C) is developed in this furnace, with the effect that slags produced are in a molten state and liquid slag is discharged off. Hence such furnaces are also called ***slagging type***.

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- **Advantages :**
  (1) Only crushed coal is used and no pulverization equipment is needed.
  (2) It removes much of the ash, about 60%, in form of molten slag. Thus only 40% ash leaves with the flue gases, compared with about 80% for pulverized coal firing.
  (3) It reduces erosion and fouling of steam generator surface due to low ash content in flue gases.
  (4) This furnace gives best results with low grade fuels like Indian coal.
  (5) It can be operated with less excess air.
  (6) High furnace temperatures are obtained.

- **Disadvantages :**
  (1) Higher forced-draft fan pressure is required therefore power requirement increases.
  (2) The formation of relatively more NO_x, which are air pollutants.
  (3) It is not suitable for high sulphur content coal.

3.12 Oil burners [Jan. '13]

The functions of the oil burner are to mix the fuel and air in the proper proportion and to prepare the fuel for combustion. Generally the oil burners are used for small capacity boilers for industrial process heating. The mixing of oil with air is done in such a way that the oil could be burnt with minimum excess air for achieving the maximum temperatures. The oil burners can be classified as follows :

(1) Vapouring burners : In these burners, the oil may be vapourised or gasified by heating within the burner. These burners are used in blow torches, gasoline stoves, etc. and not used in boilers for steam generation.

(2) Atomising burners : In these burners, the oil may be atomised by the nozzle of burner and its vapourisation occurs in combustion space. These burners are used for oil fired furnaces and boiler furnaces.

The atomisation of the oil is done in following basic ways :

(i) Steam or high pressure air atomising burners :

In these burners, the oil is broken into small droplets by using air or steam under pressure. It is also known as ***blast atomisation***. The principle of working is shown in Fig. 3.28.

The oil is usually delivered under pressure through a central tube. Steam or air is supplied along the annular area between the oil pipe and the concentric outer tube. Near the tip of the burners the two streams are brought together with resultant impingement and atomisation of fuel. Before supplying the oil to the burner, it is heated to a temperature of about 93°C to 120°C to reduce its viscosity and to improve combustion conditions. This method of atomisation is simple and cheap.

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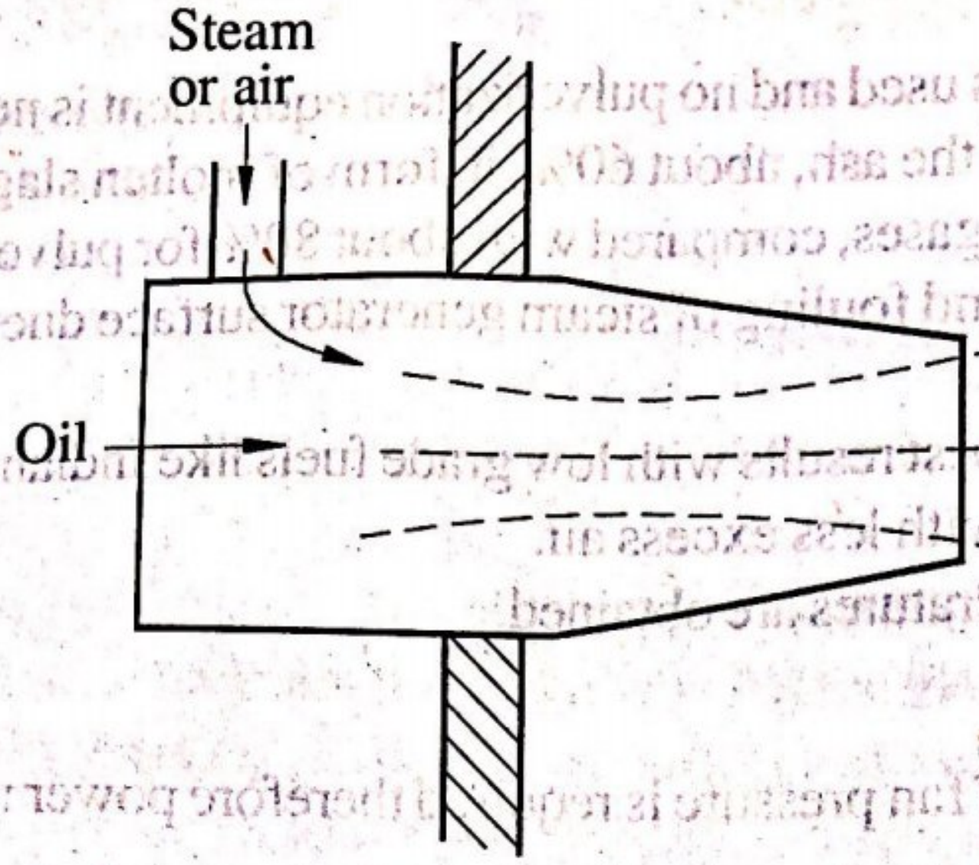


Fig. 3.28 Principle of oil firing

(ii) Mechanical burners :

In mechanical atomisation, the oil is pressurised by means of a positively displacement pump or by centrifugally dispersing the oil from a rotating wheel. ***Spray nozzle*** is used for oil

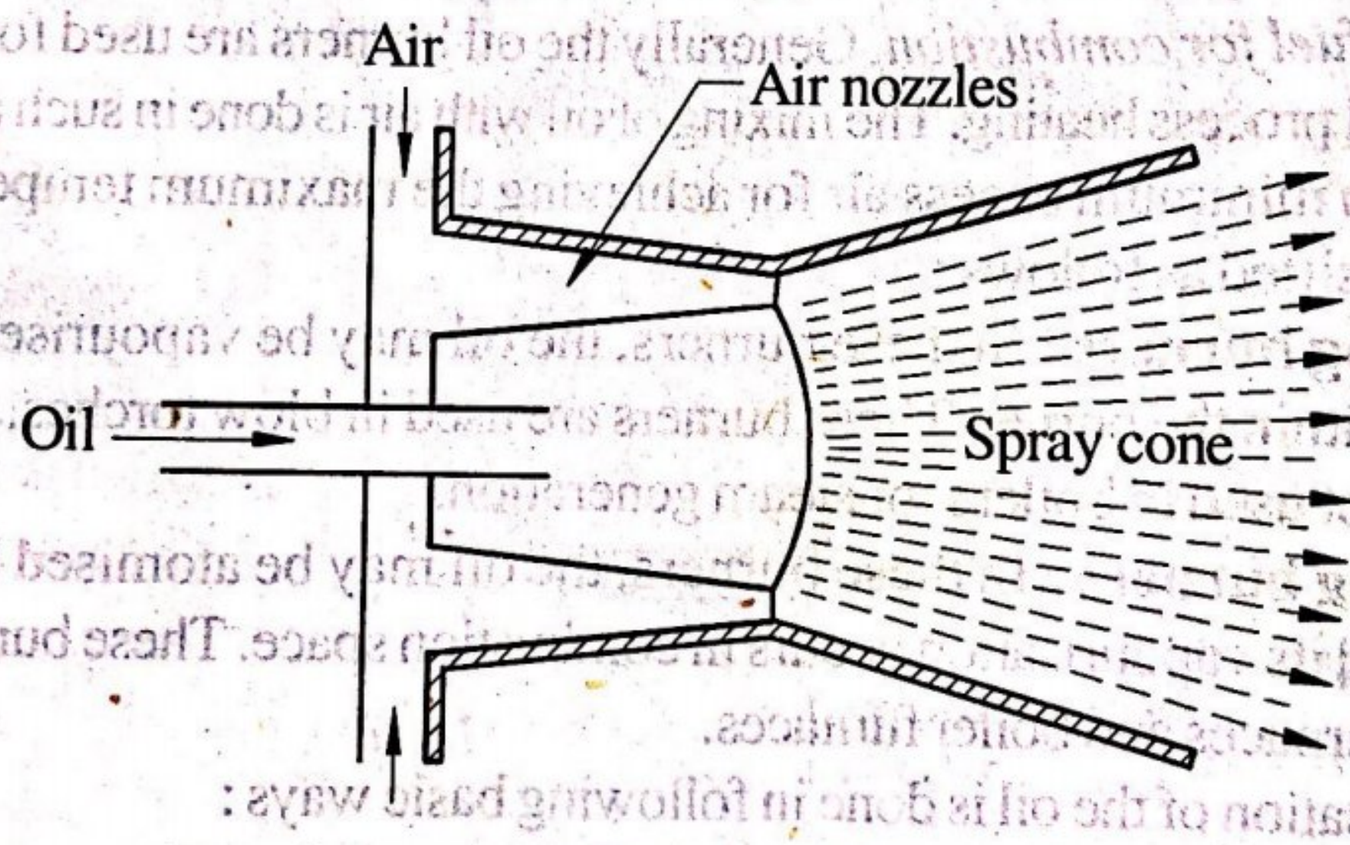


Fig. 3.29 Working of rotating wheel burner

injection under high pressure of about 25 to 35 bar. The oil leaves the nozzles in atomised form.

In case of rotating wheel burners, the oil leaves the orifice in the form of a hollow cone of fine fuel particles due to centrifugal action. The principle of working of rotating wheel burner is shown in Fig. 3.29.

The modern trend is towards mechanical atomisation since the auxiliary consumption is less, and less excess air is required.

3.13 Necessity of ash disposal

Ash is the incombustible material that remains when coal is burned. The ash came into the body of coal from the original vegetable matter from which coal was formed.

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Also they might have been ingested into coal during its formation when mineral rich water infiltrated into it.

Ash content in coal may vary from 8 to 10% to as high as 20 to 50% depending upon the grade of coal. Better the quality of coal, lower is the ash content. Hence for inferior grade coal, the ash content is higher than anthracite or semi-anthracite variety of coal. A 200 MW capacity power plant using Indian coals produces as large as 60,000 tons of ash per annum. This quantity of ash is equivalent of a height of 1.5 meters covering four hectares area. The ashes have to be discharged and dumped at a site sufficiently far away from the thermal power plant because of following reasons :

(1) The ash is dusty and irritating. Being easily airborne it causes inflammation in the eyes.
(2) It generates toxic gases and corrosive acids when it comes in contact with water.
(3) It is sufficiently hot when it comes out of the boiler furnaces.

3.14 Ash handling systems

The ash-handling is a difficult problem and sufficient attention should be given to design the ash handling plant. The following are the difficulties encountered in the ash-handling plant layout.

(1) Transportation of hot ash is difficult as well as hazardous. Hence it must be quenched before transportation.
(2) The ash produced is abrasive and will wear out the conveyor parts. Hence a special conveyor must be designed to handle the ash.
(3) It forms clinkers by fusing together in large lumps. They must be crushed to a convenient size before discharging on to a conveying equipment.

The ash handling problem is further aggravated in Indian thermal power plants using local coal due to silica content in the ash is as high as 60 to 65% which causes the erosion.

- **Ash can be utilized for**
  (1) land filling.
  (2) road making.
  (3) filling abandonded quarries.
  (4) manufacture of cement.
- **Requirements of a good ash handling plant :** **[May '13]**
  (1) It should be able to handle large quantity of clinkers, soot, boiler refuse, and dust smoothly.
  (2) The equipment should be corrosion and wear resistant.
  (3) It should be able to handle requisite quantity of ash daily.
  (4) Initial cost, operating and maintenance costs of the system should be minimum.
  (5) It should be able to handle at high rate in order to deal adequately with any sudden change in boiler operating conditions.
  (6) It should be able to operate effectively with hot and wet ashes.
  (7) It should give better efficiency under all variable load conditions.

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(8) Its operation should be noiseless as far as possible.

The general layout of the modern ash handling and dust collection plant is shown in Fig. 3.30.

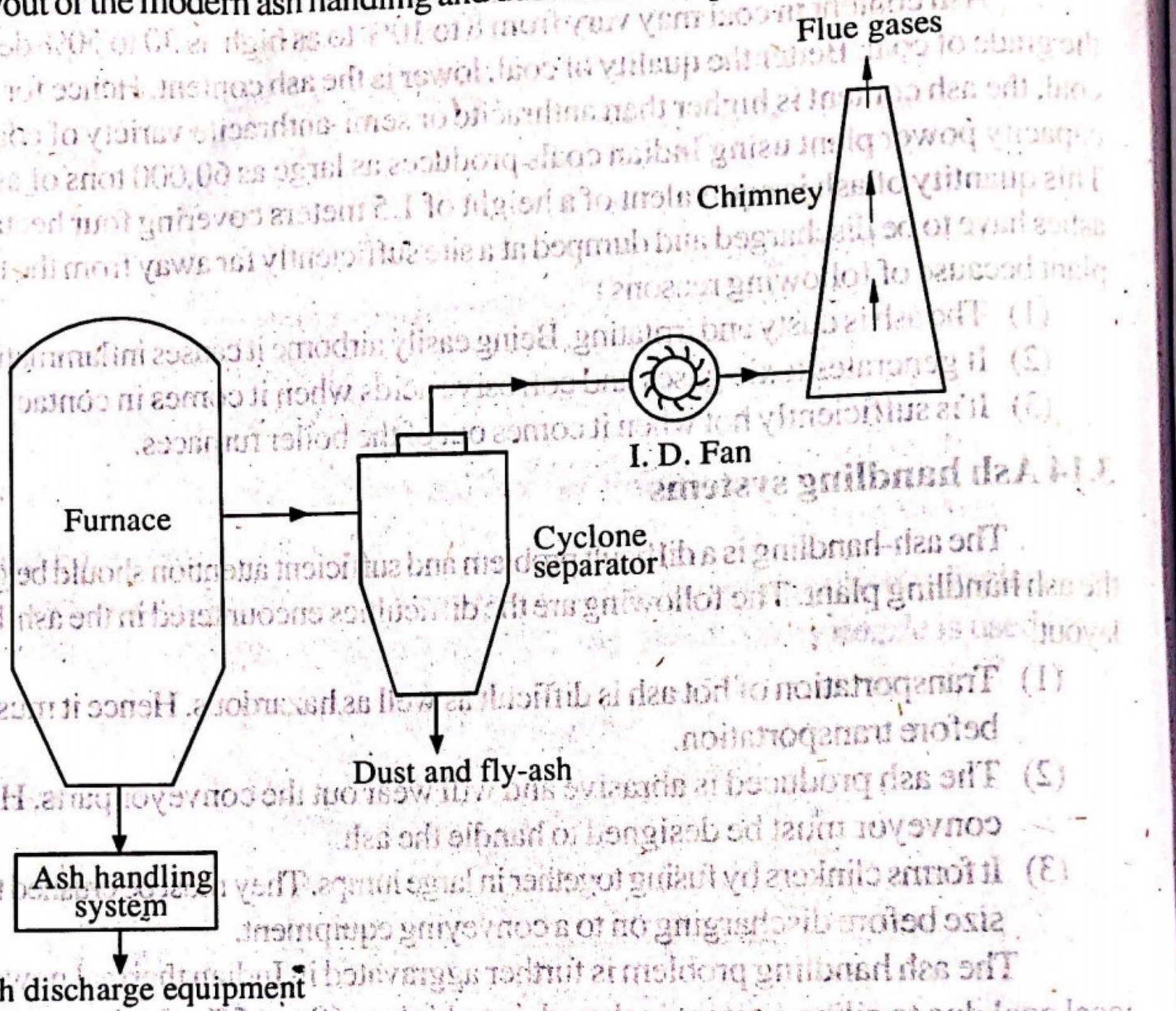


Fig. 3.30 General layout of ash handling and dust collection plant

3.15 Mechanical ash-handling system

The arrangement of mechanical ash-handling system is shown in Fig. 3.31. It is generally used for low capacity, coal fired power plants.

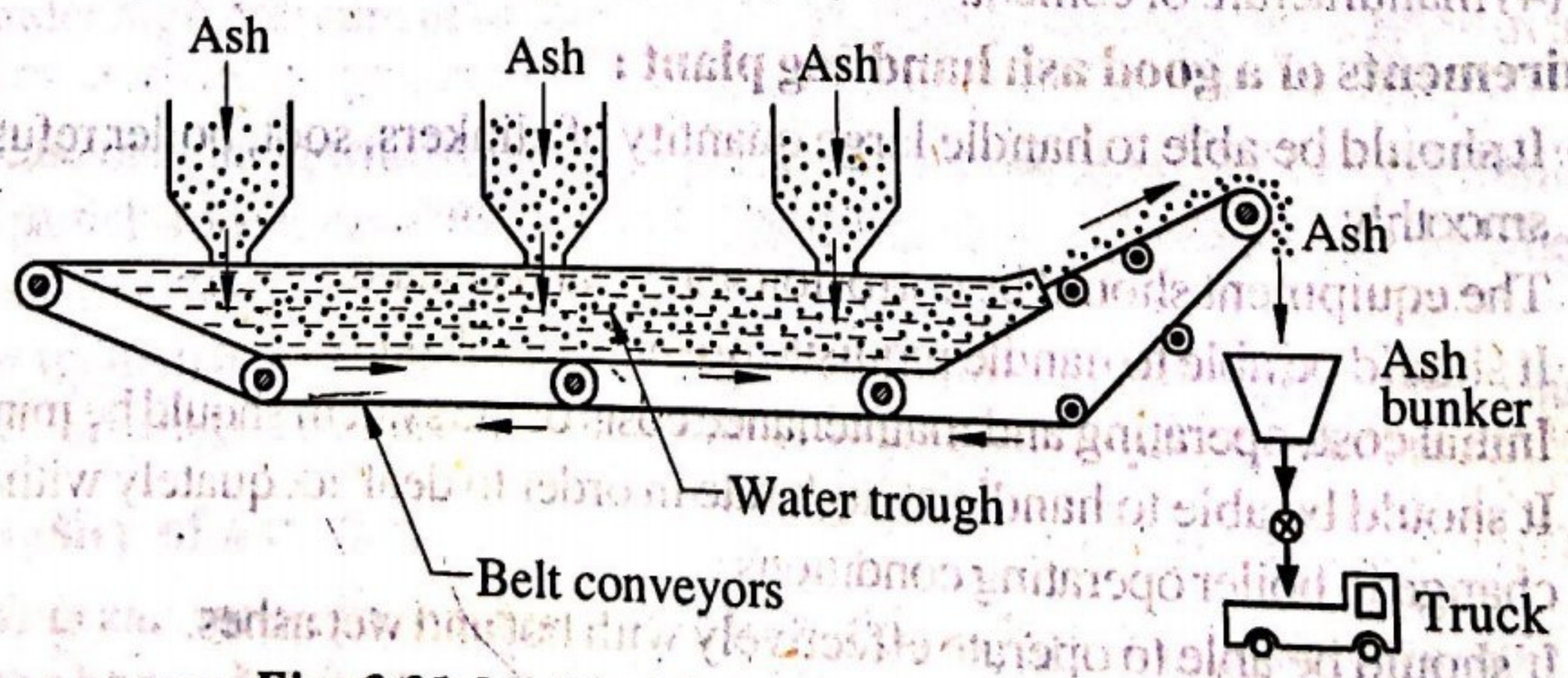


Fig. 3.31 Mechanical ash handling system

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The hot ash coming from a boiler furnace falls over a belt conveyor through a water seal. The hot ash gets cooled on quenching and is carried continuously by the belt conveyor to ash-bunker or dumping site. The ash is carried to the dumping site from the ash bunker with the help of trucks. To load the truck, control valve is operated manually.

- **Advantages :**
  (1) It has low power consumption.
  (2) Continuous ash removal.
- **Disadvantages :**
  (1) It is suitable for coal-fired, low capacity plants.
  (2) The maximum capacity of this system is limjted to 5 *t/h*.
  (3) The life span of this system is only 5 to 10 years.

3.16 Hydraulic ash-handling system

There are two types of hydraulic ash handling systems :
(1) Low-velocity system.
(2) High-velocity system.

3.16.1 Low-velocity system

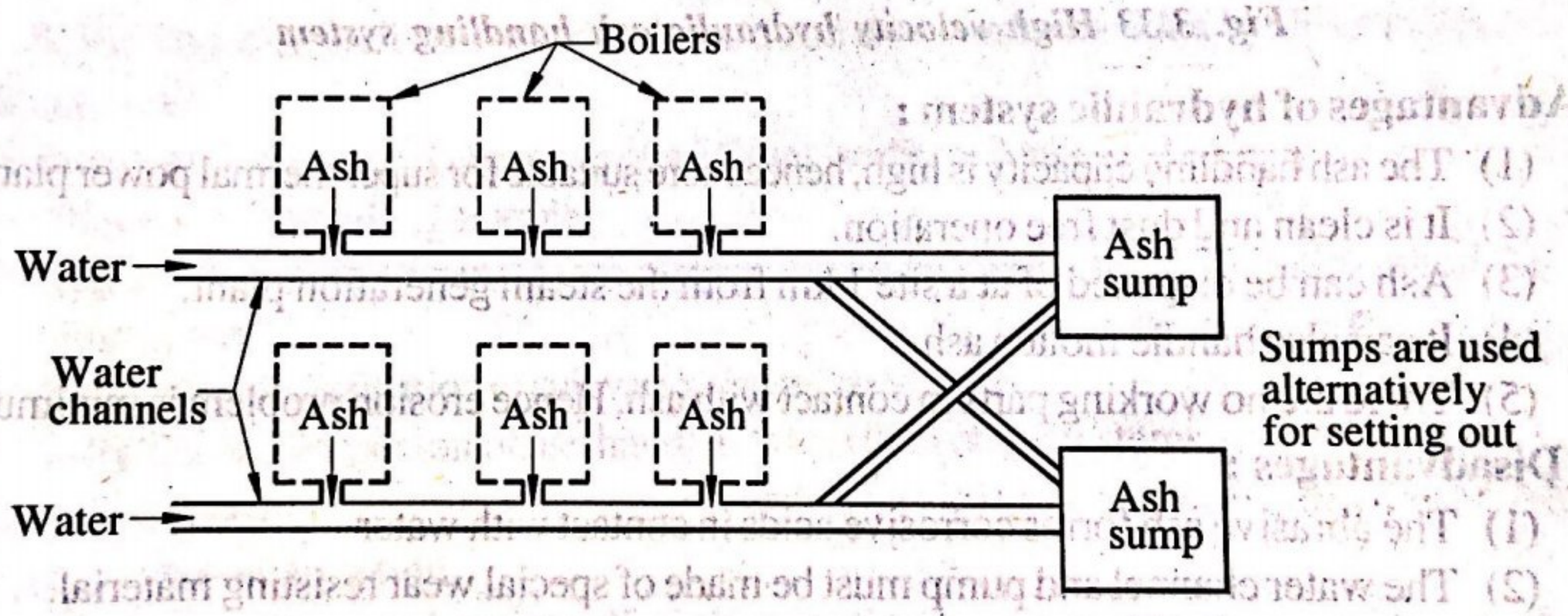


Fig. 3.32 Low-velocity hydraulic ash handling system

The arrangement of low-velocity hydraulic ash handling system is shown in Fig. 3.32. In this system, ash from the furnace grate falls into a channel of water possessing low velocity and is carried to ash sump. The velocity of water in channel is generally 3 to 5 m/s. In the sumps water and ash are separated. The ash is then loaded into trucks or wagons by means of grab buckets or belt conveyor and transferred to dump site. The separated water is used again. The ash carrying capacity of this system is 50 t/h through a distance of 500 m.

3.16.2 High-velocity system

The arrangement of high velocity hydraulic ash handling system is shown in fig. 3.33. The hoppers below the boilers are fitted with water nozzles at the top and on the sides. The

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top nozzles quench the ash while the side ones provide the driving force for the ash. The cooled ash is carried to the sump through the channel of water. The ash is allowed to settle in the ash sump and separated water is used again. The ash handling capacity of this system is as much as 120 t/h and discharged at a distance of up to 1 km.

This system can be employed in the case of a pulverized coal fired boiler also.

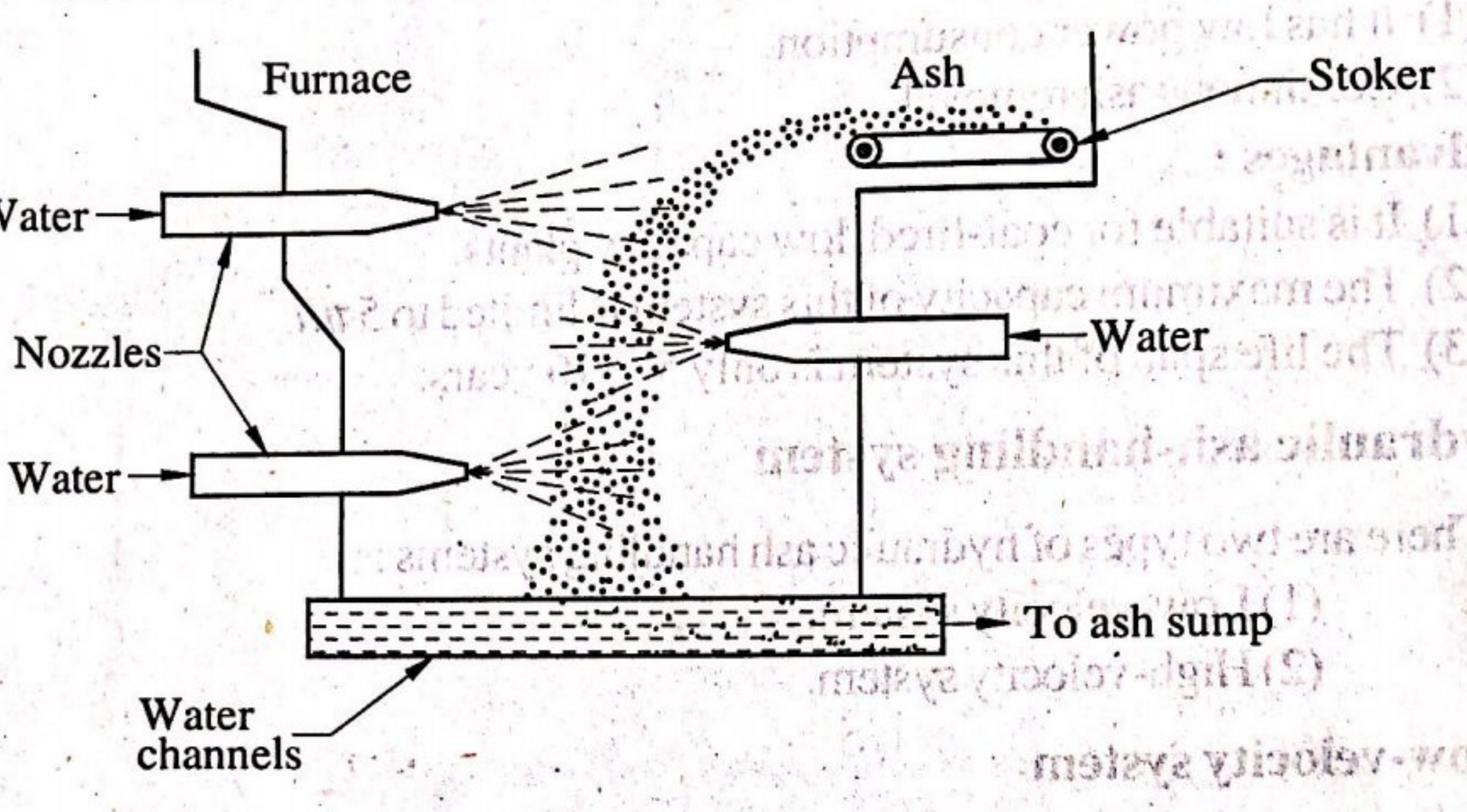


Fig. 3.33 High velocity hydraulic ash handling system

- **Advantages of hydraulic system :**
  (1) The ash handling capacity is high, hence more suitable for super thermal power plants.
  (2) It is clean and dust free operation.
  (3) Ash can be disposed of at a site 1 km from the steam generation plant.
  (4) It can also handle molten ash.
  (5) There are no working parts in contact with ash. Hence erosion problem is minimum.
- **Disadvantages :**
  (1) The abrasive ash forms corrosive acids in contact with water.
  (2) The water channel and pump must be made of special wear resisting material.

3.17 Pneumatic ash handling system [May '13, Dec. '11, Dec. '10]

The arrangement of pneumatic ash handling system is shown in Fig. 3.34. A high velocity air stream is directed to collect ash and dust from all discharge points. The ash from boiler units falls through the hopper, passes through crushers and drops into the stream of air. The ash carried by the air is separated in cyclone type separators and is collected in ash hopper. The clean air is discharged from the top of the secondary air-separator into the atmosphere through exhauster. The ash collected in the ash hopper is trucked away for disposal. The ash carrying capacity of this system varies from 5 to 30 t/h.

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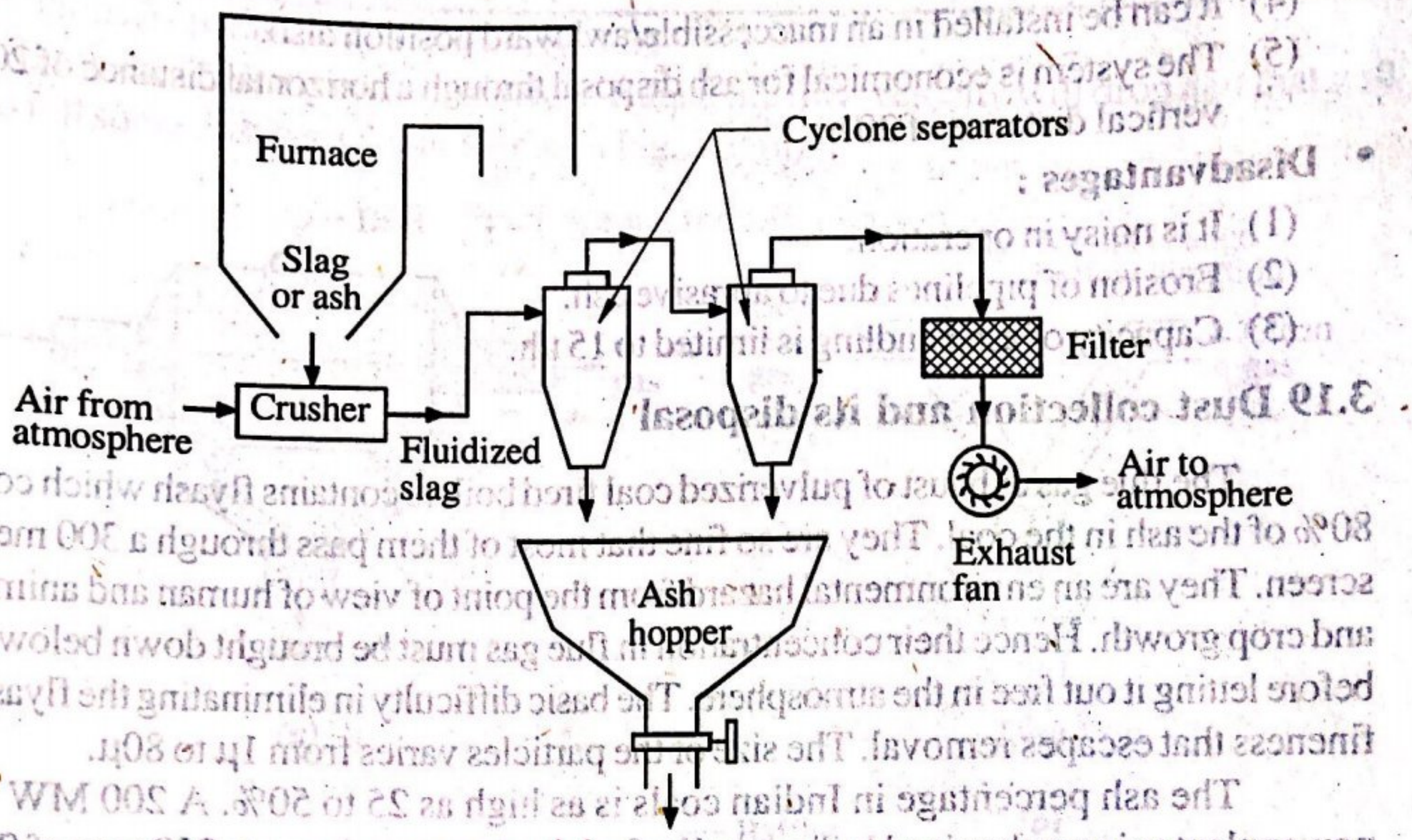


Fig. 3.34 Pneumatic ash handling system

Abrasive ash as well as finely divided ash and soot can be handled by this method.

- **Advantages :**
  - (1) It is dust free operation because of all flyash and dust is eliminated.
  - (2) There is no rehandling trouble.
  - (3) Dry ash handling eliminates the chances of ash freezing, picking or sticking in the storage bin.
  - (4) Flexibility in application to suit various plant conditions.
  - (5) Installation cost per ton of ash handled is less than other systems.
- **Disadvantages :**
  - (1) It is noisy in operation.
  - (2) Wearing of the crusher as well as the pipeline by the abrasive ash inflicts high maintenance charge.

3.18 Steam jet ash handling system [Dec. '10]

In this method of ash handling, high velocity steam instead of air is passed through a pipe to carry off dust and ash particles of considerable size. The ash is separated in the cyclone type separator and deposited in the ash hopper. The ash collected in the ash hopper is trucked away for disposal.

- **Advantages :**
  - (1) It does not require any auxiliary drive.
  - (2) Less capital cost in comparison to other systems.

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(3) Space requirement is less.
(4) It can be installed in an inaccessible/awkward position also.
(5) The system is economical for ash disposal through a horizontal distance of 200 m and vertical distance of 30 m.

- **Disadvantages :**
  (1) It is noisy in operation.
  (2) Erosion of pipelines due to abrasive ash.
  (3) Capacity of ash handling is limited to 15 t/h.

3.19 Dust collection and its disposal

The flue gas exhaust of pulverized coal fired boilers contains flyash which constitutes 80% of the ash in the coal. They are so fine that most of them pass through a 300 mesh (40μ) screen. They are an environmental hazard from the point of view of human and animal health and crop growth. Hence their concentration in flue gas must be brought down below 0.5 g/m^3 before letting it out free in the atmosphere. The basic difficulty in eliminating the flyash is their fineness that escapes removal. The size of the particles varies from 1μ to 80μ.

The ash percentage in Indian coals is as high as 25 to 50%. A 200 MW capacity power plant using pulverized Indian coal as fuel discharges as large as 250 tons of fly ash per day with the exhaust gases if it is not removed before exhaust. Therefore, it is always necessary to clean the gas streams contaminated with particulate matters before the gas is discharged to the atmosphere.

The basic requirements for a good dust collector are as follows :
(1) It must be able to remove very fine flyash particles.
(2) The operating and maintenance charges should be as low as possible.
(3) It should have low capital cost.
(4) It should be wear resistant to erosion problem inflicted by abrasive flyash.
(5) It should occupy minimum floor space.
(6) It must operate at high efficiency at all loads on the power plant.

3.20 Mechanical dust collector

The mechanical dust collector can be classified as
(1) Dry type mechanical dust collectors
  (i) Gravitational separators
  (ii) Cyclone separators
(2) Wet type mechanical dust collectors
  (i) Spray type scrubbers
  (ii) Packed bed scrubbers
  (iii) Impingement type separator

3.20.1 Gravitational separators

The gravitational separators work on the basic principle of gravity. The arrangement

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of gravitational separators is shown in Fig. 3.35.

If the dust laden gas is expanded in a duct, the flow velocity will drop and that will allow to fall some dust particles as shown in Fig. 3.35(a).

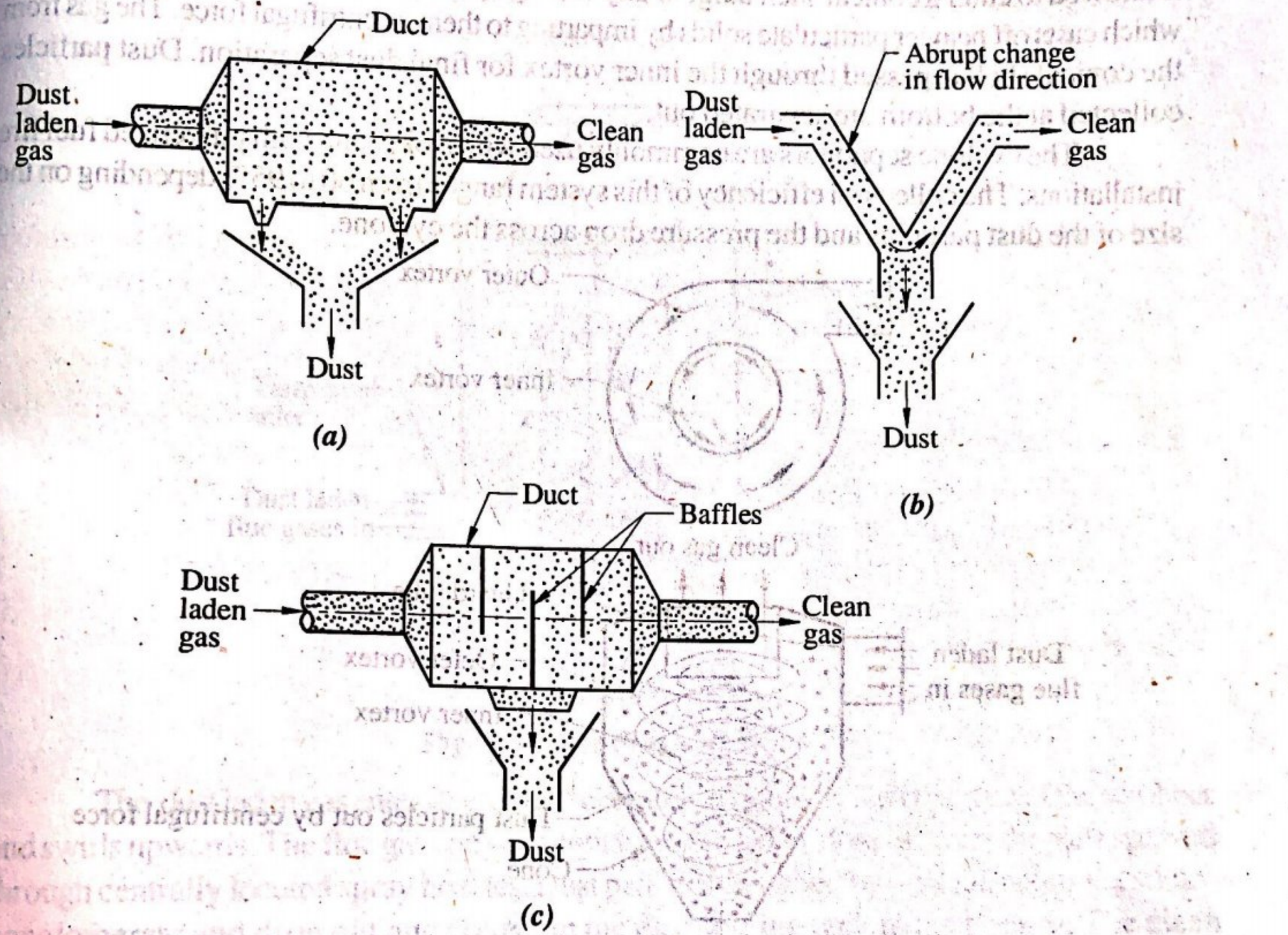


Fig. 3.35 Gravitational separators

If there is abrupt changes in the gas flow direction, it will bring about some precipitation of the heavier dust particles as shown in Fig. 3.35(b).

Placing baffles in the path of dust laden gas will bring down some heavier particulate solids due to loss of kinetic energy upon impingements shown in Fig. 3.35(c).

- **Advantages :**
  (1) Simple in arrangement.
  (2) Capital and operating costs are less.
- **Disadvantages :**
  (1) It cannot remove fine dust particles. Only large sized and heavier particles can be removed.
  (2) It is bulky and requires large space.

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3.20.2 Cyclone separator

The cyclone separator is shown in Fig. 3.36. The dust laden flue gas with high velocity is allowed to enter a conical shell tangentially setting up a swirling motion in the body of gas which casts off heavier particulate solids by imparting to them a centrifugal force. The gas from the conical shell is passed through the inner vortex for final dust separation. Dust particles collected at the bottom are separated out.

The cyclone separators are commonly used with stoker and with pulverized fuel fire installations. The collection efficiency of this system ranges from 60 to 95% depending on the size of the dust particles and the pressure drop across the cyclone.

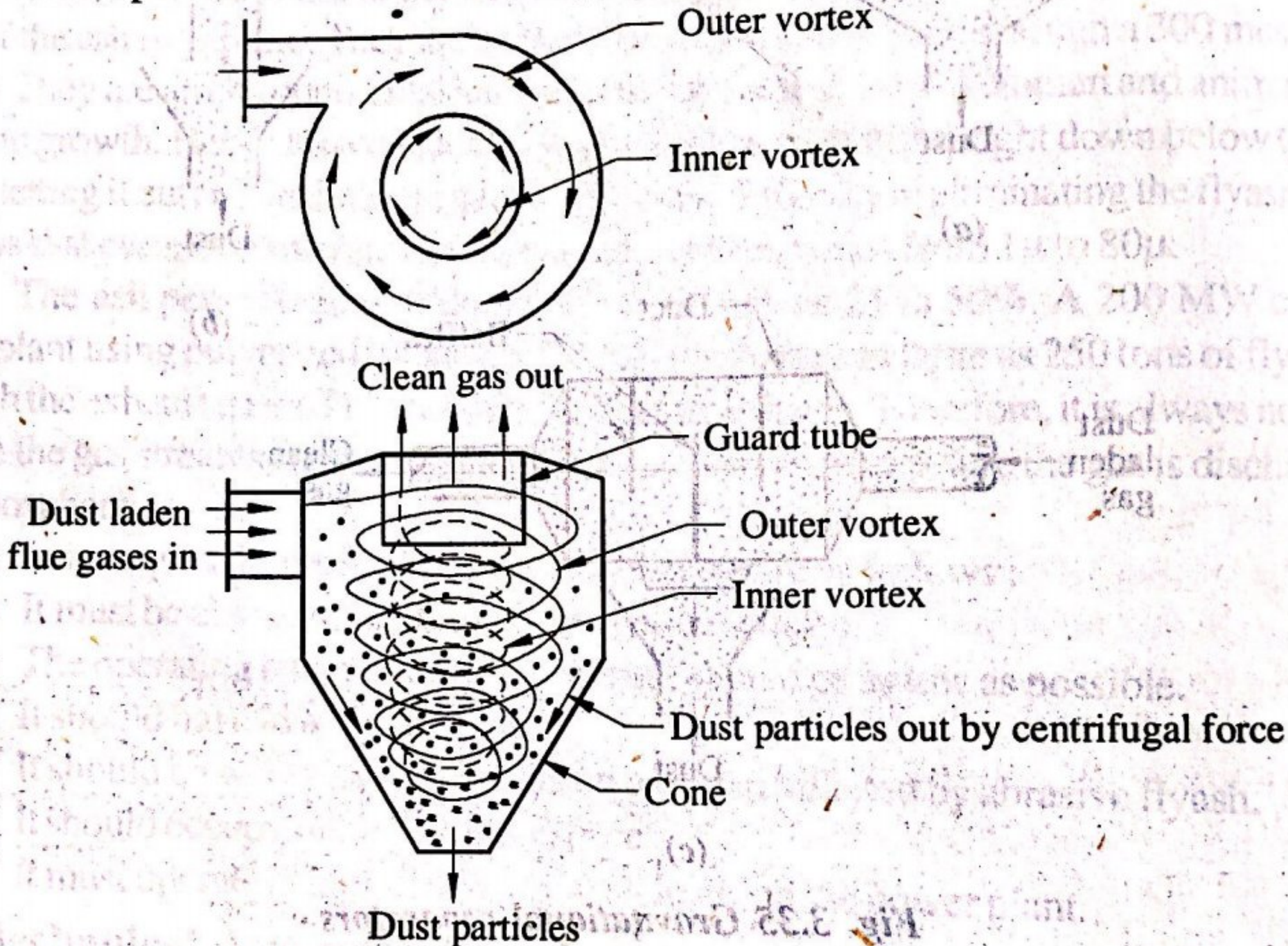


Fig. 3.36 Cyclone separator

- **Advantages :**
  (1) Efficiency is higher for bigger size particles.
  (2) Maintenance cost is low.
  (3) It is simple, cheap and easy to operate.
  (4) Its efficiency increases with increasing the load.

- **Disadvantages :**
  (1) Its efficiency declines with the increase of fineness of the particles.
  (2) It requires more power to produce very high vortex velocity of the flue gas.
  (3) It requires more space.
  (4) The pressure loss is comparatively high.

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3.20.3 Wet type mechanical dust collector

The working principle of a wet dust collector is shown in Fig. 3.37. The wet dust collector is also known as ***Gas scrubber***. The flue dust and sulphur present in the outgoing exhaust gases are washed and removed by water spraying.

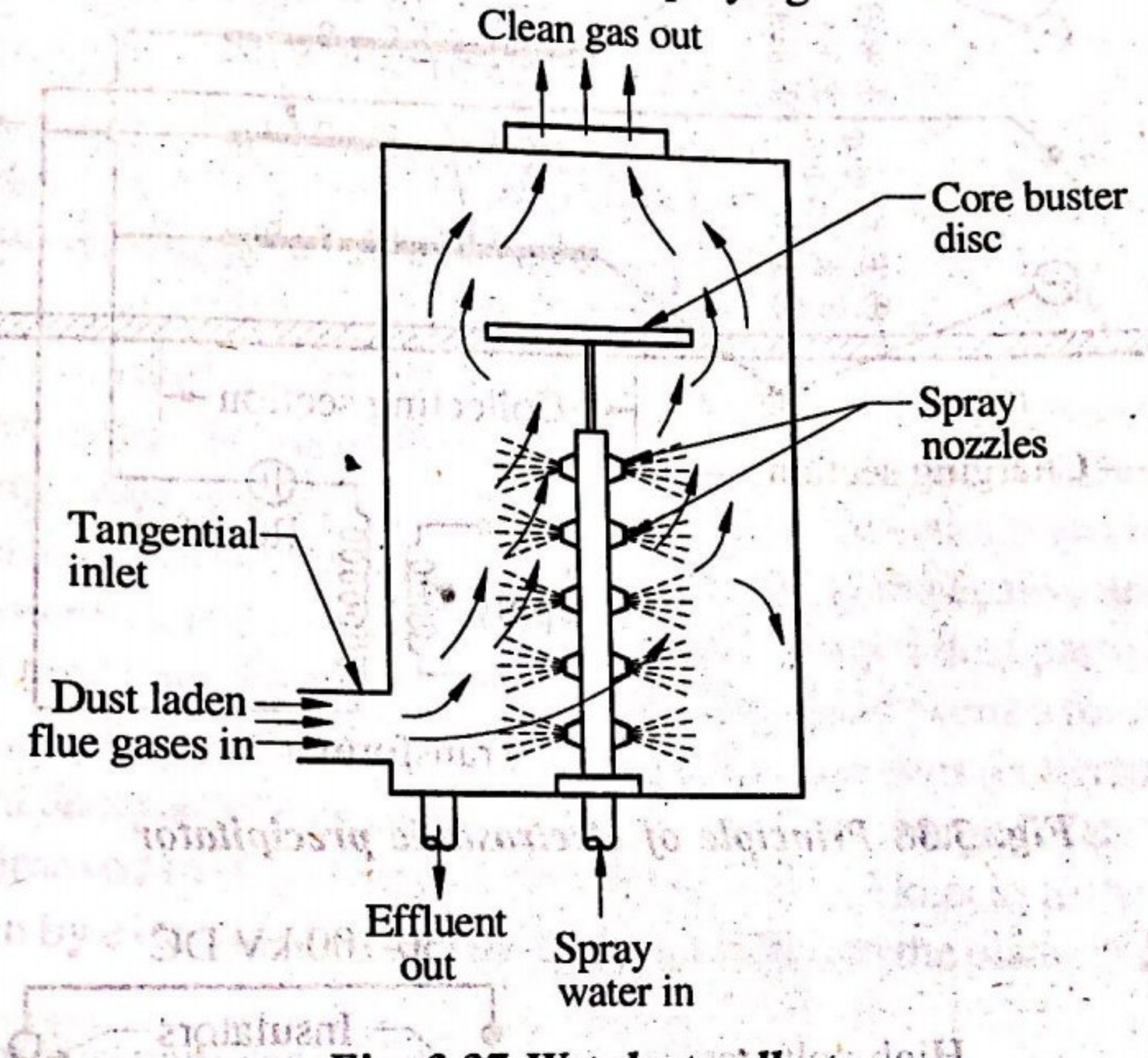


Fig. 3.37 Wet dust collector

The dust laden gas enters the vertical chamber tangentially near the base of the scrubber and swirls upwards. The flue gas comes in contact with a heavy dose of water droplets sprayed through centrally located spray nozzles. Dust particles absorbed by water droplets get sticky, conglomerate and drop out and collect in the dust settling tank at the bottom. The clean overflow liquid from the settling tank is recycled to the spray chamber.

To minimize corrosion problems, the tank is lined with lead, the pipes are lined with rubber and spray nozzles are made of vitrified materials. The efficiency of this type of dust collector is above 90% but large water consumption and corrosion problems make this method less acceptable in power plants.

3.21 Electrostatic precipitator (ESP) [Dec. '13, Jan. '13, June '11]

It is an electrical device to precipitate suspended flyash and dust particles from the flue gas by ionizing the particles in an electric field and collecting them subsequently on oppositely charged electric plates. It was first introduced by Dr. F. G. Cottel in 1906 and was first economically used in 1937 for the removal of dust and ash particles carried with the exhaust gases of the thermal power plant.

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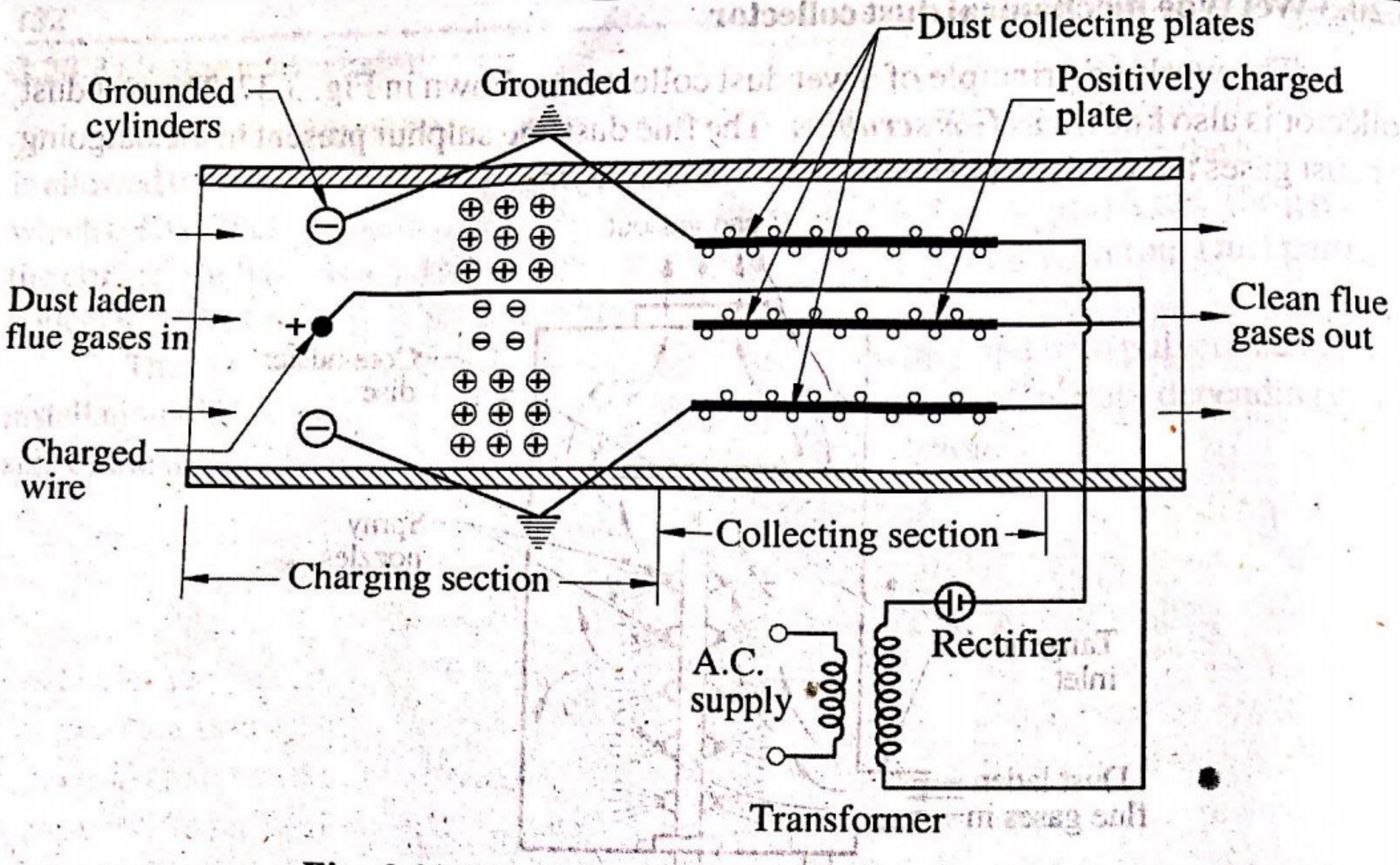


Fig. 3.38 Principle of electrostatic precipitator

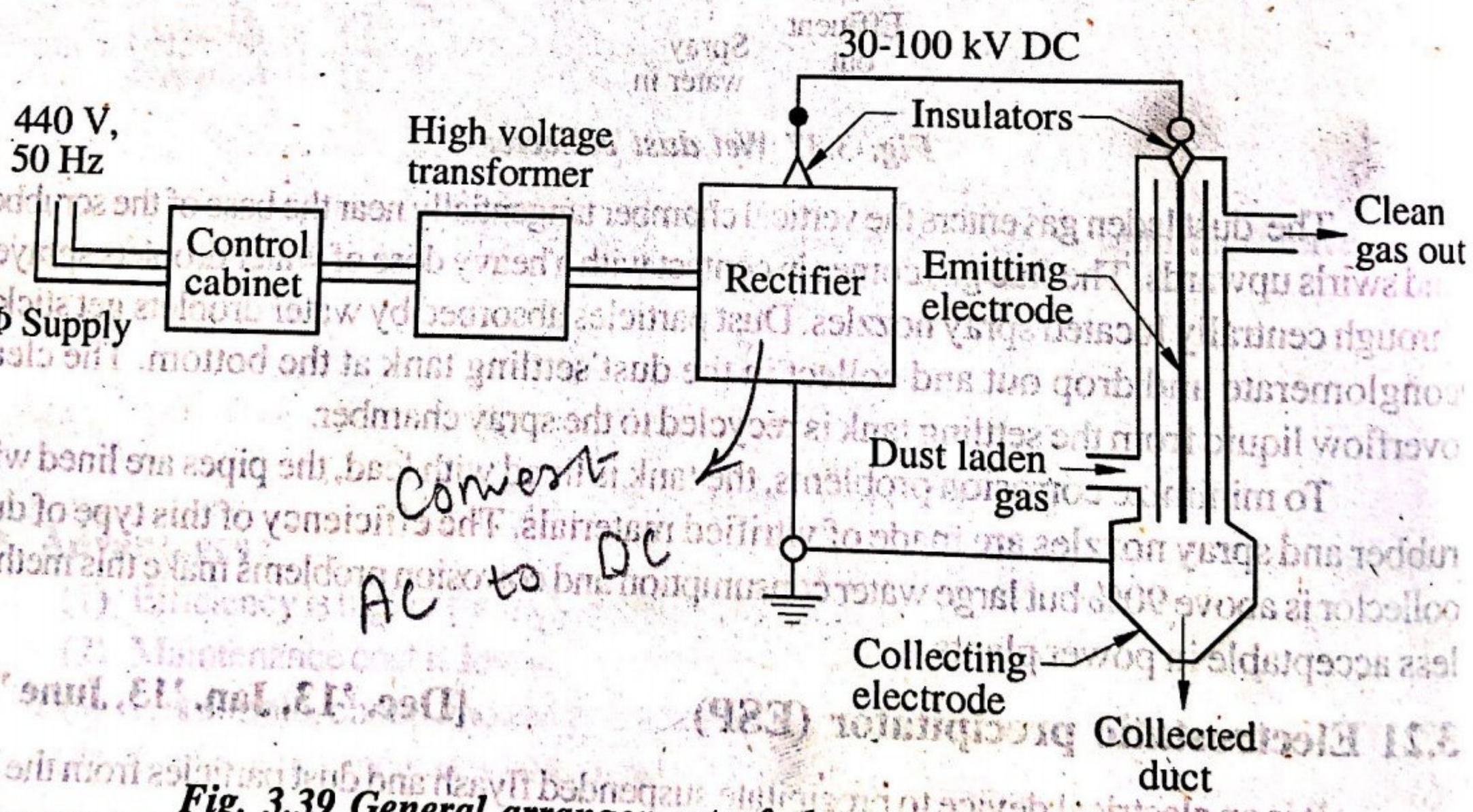


Fig. 3.39 General arrangement of electrostatic precipitator

The working principle of electrostatic precipitator is illustrated in Fig. 3.38 and general arrangement of all components is shown in Fig. 3.39. The basic components of the electrostatic precipitator are :

(1) High voltage source (440V, 50Hz, 3ϕ supply)

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(2) Ionizing and collecting electrodes
(3) High voltage transformer
(4) Rectifier to convert AC into DC.

The two sets of electrodes (ionizing and collecting) are insulated from each other. The first set consists of rows of electrically grounded vertical parallel plates, called the ***collection electrodes***. The second set of electrodes is composed of wires, called the ***discharge*** or ***emitting*** electrodes that are centrally located. The emitting electrodes (wires) carry a unidirectional positively charged high-voltage (between 30 to 100 kV) current from an external DC source.

When the dust laden gas is passed between charged wires and grounded cylinders this gas becomes ionised as the voltage applied between the electrodes is as high as 30,000 to 100,000 V. The magnitude of applied voltage is dependent on electrode spacing. As the dust laden gas is passed through these highly charged electrodes, both negative and positive ions are formed, the latter being as high as 80%. When the charged dust particles are passed between the plates (collecting electrodes), the electrostatic field exerts a force on positively charged dust particles and drives them towards the grounded plates (collecting electrodes). Negatively charged dust particles attract towards the positively charged plate. The deposited dust particles are removed from the plates by giving the vibration knocks to the plates with the help of cams driven by external means. The removed dust from the plates is collected in the dust hoppers.

If the size of dust particles is high, ESP is used in combination with cyclone separator. The ESP has a high collection efficiency of the order of 90% and above.

The electrostatic precipitators are extensively used for removing flyash from flue gas in coal fired thermal power plants.

- **Advantages :**
  (1) It is the most effective method to remove very fine particulates, as fine as 0.01 μ, which escape removal by mechanical separators.
  (2) It is very useful for high dust loaded gas. Flue gas containing dust particles as high as 100 g/m^3 can be effectively cleaned.
  (3) Dust removal efficiency is as high as 99.5%.
  (4) Its operation is easy and smooth.
  (5) The draught loss (pressure loss) is minimum.
  (6) It can remove both wet and dry dust.
- **Disadvantages :**
  (1) Its capital cost is high.
  (2) The running charges are high as the amount of power required for charging is considerably large.
  (3) Its removal efficiency drops with the increase of gas velocity.
  (4) It requires more floor space.

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Exercise

3.1 Why is coal-handling system called lifeline of the coal fired power plants ?
3.2 What do you understand by out-plant handling of coal ?
3.3 Explain different modes of out-plant handling of coal with limitations.
3.4 Why is coal transportation by road suitable only for small capacity plants ?
3.5 What are the advantages and disadvantages of coal transportation by pipeline?
3.6 Why is coal storage required ?
3.7 Justify : the large amount of coal storage is undesirable.
3.8 What is dead storage of coal ? Explain different methods of it.
3.9 What do you mean by live storage of coal ?
3.10 Why mechanical handling of fuel is necessary in power plants ?
3.11 Explain the various steps involved in in-plant coal handling.
3.12 Explain the following systems used in coal transfer :
(1) Belt conveyor (4) Grab bucket elevator
(2) Screw conveyor (5) Skip hoist, and
(3) Bucket elevators (6) Flight conveyor
3.13 Explain in-plant handling of coal with neat sketch.
3.14 Explain various outplant handling and inplant handling systems for coal in brief.
3.15 Give advantages and disadvantages of stoker firing.
3.16 Explain the function of hot primary air, secondary air and tertiary air in combustion of fuel.
3.17 What is meant by "overfeed" and "underfeed" principle of coal firing. Give the advantages and disadvantages of each.
3.18 Explain the traveling grate stoker with the help of neat sketch.
3.19 Explain the spreader stoker with the help of neat sketch.
3.20 Explain the following underfeed stokers :
(1) single retort stoker, (2) Multi retort stoker.
3.21 What are the advantages and disadvantages of pulverized coal firing.
3.22 Explain the two stages of coal pulverization.
3.23 Why is pulverized coal used in power plants ? Explain its advantages.
3.24 Compare pulverized coal firing with stoker firing.
3.25 Explain unit system of pulverized fuel handling with neat sketch.
3.26 Explain bin (central) system of pulverized fuel handling with neat sketch.
3.27 Write short note on ball mill for coal pulverizing.
3.28 Write short note on Ball and race type coal pulverizing mill.
3.29 Explain bowl mill of coal pulverization with neat sketch.
3.30 Explain hammer mill of coal pulverization with advantages and disadvantages.
3.31 Explain the following pulverized coal burners :
(1) Long flame burner (2) Short flame burner (3) Tangential burner

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3.32 Write short note on cyclone burner.
3.33 Discuss cyclone burner used for pulverized fuel with neat sketch.
3.34 Explain slagging type coal fired burner with advantages and disadvantages.
3.35 Explain mechanical type oil burner with neat sketch.
3.36 Explain necessity of ash disposal.
3.37 State requirements of a good ash handling plant.
3.38 Classify various ash handling system. Explain pneumatic and steam jet ash handling system.
3.39 Explain mechanical ash-handling system with neat sketch.
3.40 Explain the pneumatic ash handling system with sketch.
3.41 Explain low-velocity and high velocity hydraulic ash handling system with neat sketches.
3.42 Why dust collection and its disposal is required in thermal power plant ?
3.43 Explain dry type mechanical dust collectors with neat sketches.
3.44 Give the classification of dust collectors and explain wet dust collector with sketch.
3.45 Explain the working of electrostatic precipitator with neat sketch.
3.46 Write short note on electrostatic precipitator with sketch.

Objective Type Questions

1. The transportation of coal from _____ is called outplant coal handling.
   (a) dead storage to live storage (b) mines to power plant
   (c) live storage to boiler furnace (d) dead storage to coal preparation plant
2. In India, the coal supply to power plants is mainly by following mode of transportation.
   (a) waterways (b) highways (c) railways (d) ropeways
3. The following period of coal storage is more advantageous :
   (a) 15 to 30 days (b) 30 to 90 days
   (c) 90 to 120 days (d) 120 to 150 days
4. Coal preparation can be achieved by
   (a) crusher (b) driers (c) magnetic separator (d) all of the above
5. The following is not a characteristic of belt conveyors
   (a) Its maintenance cost is high.
   (b) It is suitable for medium and large power plants.
   (c) It is not suitable for short distances.
   (d) Its speed varies from 1 m/s to 1.8 m/s.
6. The following conveyor is extensively used for vertical lifts
   (a) belt conveyor (b) screw conveyor
   (c) bucket elevators (d) flight conveyor
7. _____ method of firing is more suitable for coal having high volatile matter.
   (a) underfeed stokers (b) overfeed stokers
   (c) hand firing (d) all of the above

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8. _____ air is supplied separately to complete the combustion.
(a) primary (b) secondary
(c) tertiary (d) none of the above

9. The following is not characteristic of pulverized coal firing :
(a) Pulverization brings about a large increase in surface area per unit mass of coal.
(b) less excess air is required for complete combustion.
(c) Pulverized coal burns on grate.
(d) It can be used for large capacity power plant.

10. The following statement is not true for ball mill.
(a) It is one of the oldest pulverizing mill. (b) It is simple in operation and low in initial cost.
(c) It is larger and heavier in construction. (d) Its operating cost is less.

11. The following statement is true for ball and race mill :
(a) It has low power consumption.
(b) It works on the principle of impact.
(c) It is not suitable for variable load.
(d) It is higher in weight.

12. Long flame burner is suitable for
(a) high volatile coal (b) low volatile coal
(c) oil burning (d) all of the above

13. The following is called slagging type burner :
(a) Long-flame burner (b) Turbulent burner
(c) Tangential burner (d) cyclone burner

14. The following burner can use crushed coal for firing.
(a) Long-flame burner (b) Turbulent burner
(c) Tangential burner (d) cyclone burner

15. If the silica content of ash is higher, what trouble will originate ?
(a) the greater is the corrosion problem.
(b) the greater is the erosion problem.
(c) the greater is the erosion and corrosion problem.
(d) generates toxic gases.

16. The ash can be utilized for
(a) land filling (b) road making
(c) filling abandonded quarries (d) all of the above

17. What is the main limitation of the mechanical ash handling system ?
(a) It has high power consumption.
(b) It can not remove ash continuously.
(c) It is suitable for low capacity coal fired plants only.
(d) all of the above.

18. Erosion problem is minimum in _____ ash handling system.

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(a) mechanical (b) pneumatic (c) hydraulic (d) steam-jet

19. The size of the fly-ash particles varies from _____
(a) 1 to 80 μ (b) 80 μ to 100μ (c) 100 μ to 150 μ (d) 150 μ to 200 μ

20. The following statement is not true for electrostatic precipitator :
(a) Its capital cost is high.
(b) Its dust removal efficiency drops with the increase of gas velocity.
(c) It is not suitable for high dust loaded gas.
(d) It can remove both wet and dry dust.

Answers

1. (b)	**2.** (c)	**3.** (b)	**4.** (d)	**5.** (a)	**6.** (c)	**7.** (a)	**8.** (b)	**9.** (c)	**10.** (d)
11. (a)	**12.** (b)	**13.** (d)	**14.** (d)	**15.** (b)	**16.** (d)	**17.** (c)	**18.** (c)	**19.** (a)	**20.** (c)

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GTU Study Materials ☑

Chapter 4
Draught System

4.1 Introduction to draught
4.2 Losses in the air-gas circuit of boiler
4.3 Classification of draught
4.4 Natural draught
4.4.1 Estimation of height of chimney
4.4.2 Draught in terms of hot gases column
4.4.3 Diameter of chimney
4.4.4 Condition for maximum discharge through a chimney
4.4.5 Draught at maximum discharge
4.4.6 Efficiency of chimney
4.4.7 Advantages and limitations of chimney or natural draught
4.5 Mechanical draught
4.5.1 Forced draught
4.5.2 Induced draught
4.5.3 Balanced draught
4.5.4 Comparision between forced and induced draught
4.5.5 Power requirement by fans
4.6 Advantages and disadvantages of mechanical draught over natural draught
4.7 Steam jet draught

4.1 Introduction to draught [Dec. '11]

In a boiler the combustion of the fuel requires to supply sufficient amount of air and removal of exhaust gases and this is taken place by draught system.

Draught is defined as ***"the pressure difference (between absolute gas pressure at any point in a gas flow passage and the atmospheric air outside the boiler) required to maintain flow of air and exhaust gases through the boiler setting."***

Function of draught :

(i) To supply required quantity of air to the boiler furnace for the combustion of fuel.
(ii) To remove exhaust gases from the boiler passage to maintain flow of air and gases.
(iii) To create pressure difference against pressure losses in the flow passage.
(iv) To discharge the exhaust gases such that they will not objectionable or injurious to the surroundings.

4.2 Losses in the air-gas circuit of boiler

As we know that draught is nothing but pressure difference required to necessary to

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accelerate the burnt gases to their final velocity and to overcome the pressure losses in the boiler passage.

Draught = [head required to accelerate the gases] + [head losses in the path of flue gases]

$$h_t = [h_v] + [h_a + h_g + h_d]$$

where h_t = total draught required or total loss

h_v = pressure loss due to gas exit velocity from the chimney

h_a = pressure loss on air side

= sum of losses in air side as air preheater, damper, air ducts, and furnace bed.

h_g = pressure loss on gas side

= sum of losses in gases path as superheater, reheater, economizer, gas side air preheater, and dust collectors

h_d = pressure losses in chimney and gas ducts due to friction.

Pressure loss in furnace bed or fuel resistance : The pressure loss in furnace bed depends on fuel size, bed thickness and combustion rate.

Pressure losses due to equipment in air and gas sides : These losses depends on the size and obstruction provided in the flow path. The manufacturers usually provide data for equipments like superheater, reheater, air preheater, economizer etc. These losses follows a parabolic law.

Pressure loss in chimney and gas ducts : These loss depends on co-efficient of friction of surface of duct and chimney. The frictional loss is given by $h_d = \frac{4flV^2}{D \times 2g}$ where f = co-efficient of friction, l = length of duct, V = velocity of gas, D = diameter of duct.

Velocity head loss (h_v) : The flue gases leaving the chimney, it requires some amount of velocity to diffuse and mix with the atmospheric air. This loss is given by $\frac{V^2}{2g}$, where V velocity of flue gases leaving the chimney. The velocity head depends on the air velocity of outside surroundings. The velocity head require at outlet of chimney is larger when the velocity of outside air is higher.

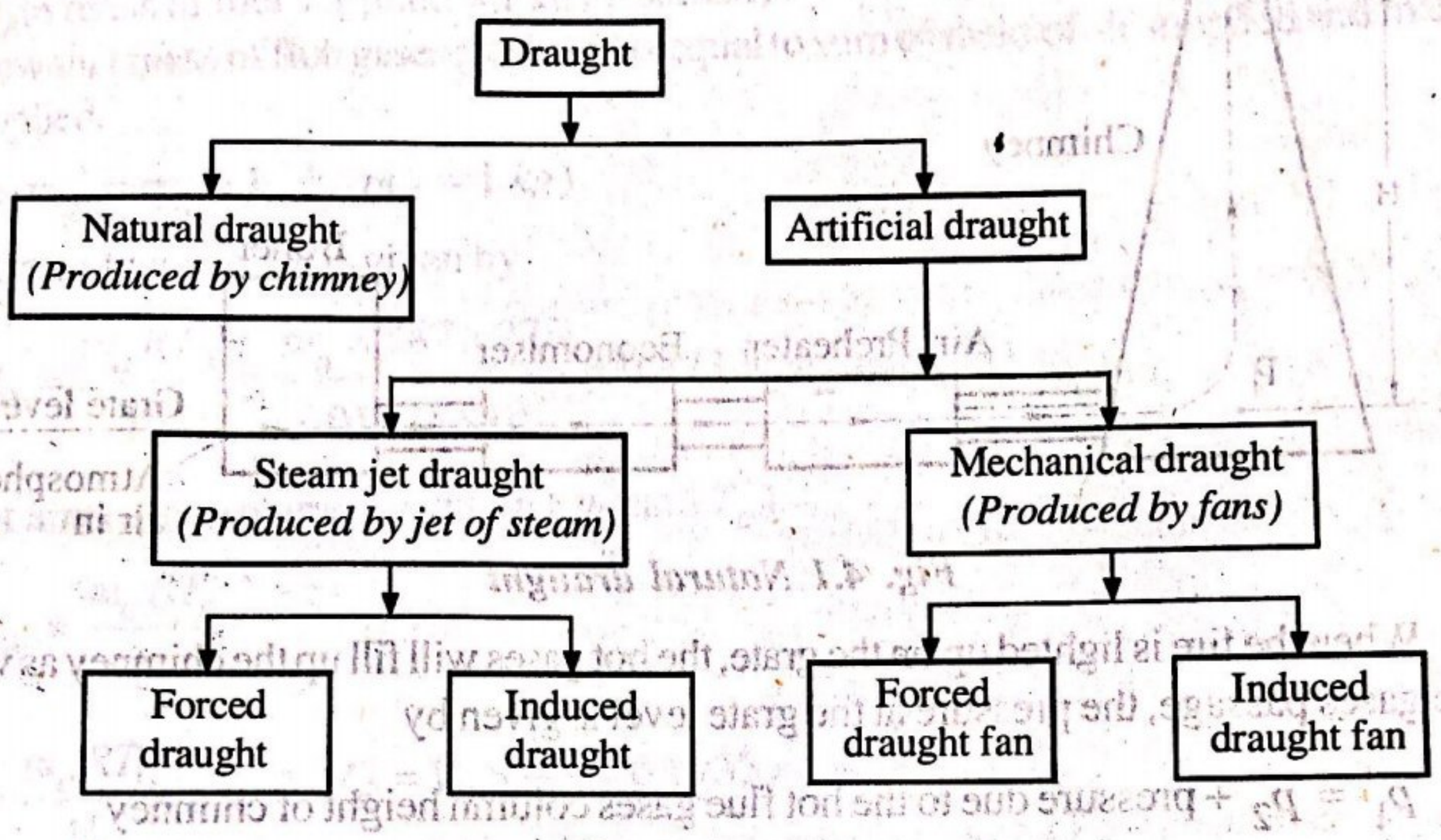
The various draught losses in different parts of boiler are measured in mm of water with help of manometers. To get accurate measurement inclined type manometer is used in which one end opened to atmosphere and other end is connected to the point in the air/gas passage where draught is to be measured.

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4.3 Classification of draught

The detail classification of draught is as follows :



4.4 Natural draught

The natural draught is obtained by the use of chimney. The chimney in the power plant performs the following two functions.

(i) It produces draught (create pressure difference) which required to force air and gases through the furnace, boiler accessaries and settings.

(ii) It carries the product of combustion to such a height that they will not be objectionable or injurious to surrounding.

The chimney is vertical tabular structure build either of masonary, concrete or steel. The draught produces by the chimney is due to density difference between the column of hot gases inside the chimney and cold air outside the chimney and also on the height of the chimney above the level of the furnace grate. The density difference depends on the temperature of flue gases passing in chimney and temperature of atmospheric air.

Consider, a chimney height H above grate level as shown in Fig. 4.1.

Let, p_1 = pressure at grate level (chimney base)

p_2 = pressure at top of chimney

When fire is not lighted, the chimney is filled by atmospheric air, the pressure at grate level is given by

$p_1 = p_2$ + pressure due to atmospheric air column of height H

$$\therefore p_1 = p_a + \rho_a g H \quad \ldots(4.1)$$

where p_a = atmospheric air pressure, ρ_a = density of atmospheric air

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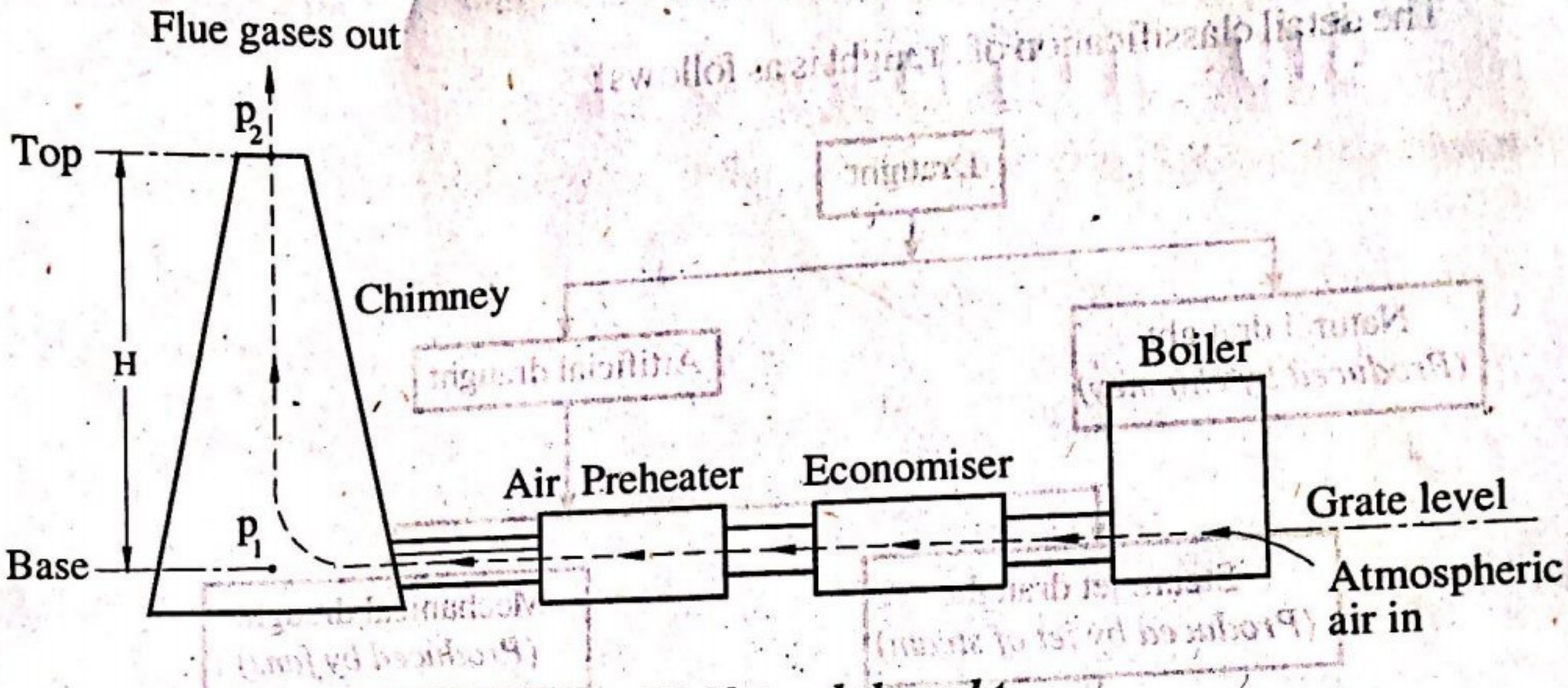


Fig. 4.1 Natural draught

When the fire is lighted up on the grate, the hot gases will fill up the chimney as well as the flue gases passage, the pressure at the grate level is given by

$p_1' = p_2 +$ pressure due to the hot flue gases column height of chimney

$$\therefore p_1' = p_a + \rho_g g H \quad ...(4.2)$$

where, ρ_g = density of hot flue gases

Now, net pressure difference at the grate level

$$\Delta p = p_1 - p_1'$$

$$= (p_a + \rho_a g H) - (p_a + \rho_g g H)$$

$$\therefore \Delta p = (\rho_a - \rho_g) g H \quad ...(4.3)$$

The density of hot flue gases is less than the density of atmospheric air, hence Δp positive which causes the flow of gases. The pressure difference Δp, causing the flow of gases is called as ***static draught.***

4.4.1 Estimation of Height of Chimney [Dec. '13, Jan. '13, June '11]

The amount of natural draught produced by a chimney mainly depends on height of chimney, temperature of hot gases and atmospheric air.

Let, T_0 = absolute temperature corresponding to 0°C

T_a = absolute temperature of atmospheric air, K.

T_g = absolute temperature of flue gases inside the chimney, K

H = height of chimney, m

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m_a = mass of air supplied to boiler (kg per kg of fuel supplied)

m_g = mass of flue gases (kg/kg of fuel)

m_f = mass of fuel supplied = 1 kg (assumed)

As we know that mass of flue gases produced is equal to sum of mass of air supplied and mass of fuel supplied.

$\therefore m_g = m_a + 1 \quad (\because m_f = 1\,kg)$

Now, volume of air at NTP is given by

$$V_0 = \frac{m_a R T_0}{p_0} = \frac{m_a \times 287 \times 273}{1.01325 \times 10^5} = 0.77326\, m_a \approx 0.7733\, m_a$$

Volume of air at atmospheric condition (p_0 and T_a)

$$V_a = \frac{m_a R T_a}{p_0}$$

But $p_0 = \frac{m_a R T_0}{V_0}$ $\quad \therefore V_a = V_0 \times \frac{T_a}{T_0} = 0.7733\, m_a \times \frac{T_a}{273}$

Similarly, volume of flue gases at p_0 and T_g

$$\therefore V_g = 0.7733 m_a \times \frac{T_g}{273}, \quad \text{(neglecting mass of fuel)}$$

Also, density of air $\rho_a = \frac{m_a}{V_a} = \frac{m_a}{0.7733 m_a} \times \frac{273}{T_a} = \frac{353}{T_a}$

density of flue gases $\rho_g = \frac{m_g}{V_g} = \frac{(m_a + 1)}{0.7733 m_a} \times \frac{273}{T_g} = \left(\frac{m_a + 1}{m_a}\right) \times \frac{353}{T_g}$

The value of ρ_a and ρ_g putting in equation (4.3), we get total static pressure difference Δp in N/m^2.

$$\Delta p = 353 g H \left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a + 1}{m_a}\right)\right] N/m^2 \qquad ...(4.4)$$

The pressure difference Δp can be expressed in terms of mm of water column (h_w).

$$\Delta p = \rho_w g h_w$$

$$\therefore h_w = \frac{\Delta p}{\rho_w g} \text{ m of water column} = \frac{\Delta p}{1000 \times g} = \frac{\Delta p}{g} mm \text{ of water column}$$

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$$\therefore h_w = 353\,H\left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right)\right],\ mm \text{ of water column}$$

Thus, we see that the height of a chimney determines the draught which it will develop under the given conditions of atmospheric air and flue gas temperature.

The Δp can be increased either by increasing the height of the chimney or allowing the flue gases to go out of boiler at higher temperature.

4.4.2 Draught in terms of hot gases column

Assuming that the draught Δp produced is equivalent to H_1 **metre height of burnt flue gases**

$$\therefore \Delta p = \rho_g g H_1 = \left(\frac{m_a+1}{m_a}\right) \times \frac{353}{T_g} \times g H_1 \quad \ldots(4.5)$$

From equations (4.4) and (4.5), we get

$$353 g H\left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right)\right] = \left(\frac{m_a+1}{m_a}\right) \times \frac{353}{T_g} \times g H_1$$

$$\therefore \frac{H}{T_a} - \frac{H}{T_g}\left(\frac{m_a+1}{m_a}\right) = \frac{H_1}{T_g}\left(\frac{m_a+1}{m_a}\right)$$

$$\therefore H_1\left(\frac{m_a+1}{m_a}\right) = H \times \left[\frac{T_g}{T_a} - \left(\frac{m_a+1}{m_a}\right)\right]$$

$$\therefore H_1 = H \times \left[\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a} - 1\right] m \text{ of flue gases column} \quad \ldots(4.6)$$

4.4.3 Diameter of chimney

The mass of the gases flowing through any cross section of the chimney is given by

$$\dot{m}_g = \rho_g A.C,\ kg/s$$

where, $A = \frac{\pi}{4}D^2$, D = diameter of chimney at any cross section

C = velocity of flue gases passing through chimney.

$$\therefore D = 1.128\sqrt{\frac{\dot{m}_g}{\rho_g . C}} \quad \ldots(4.7)$$

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The velocity of gases passing through chimney is given by

$$C = \sqrt{2g(H_1 - h')}$$

where h' = head loss in the chimney in terms of metre of flue gases column.

$$\therefore C = 4.43\sqrt{H_1 - h'} = 4.43\sqrt{H_1}\sqrt{1 - \frac{h'}{H_1}} = K\sqrt{H_1}$$

where $K = 4.43\sqrt{1 - \frac{h'}{H_1}}$ = constant

$K = 0.825$ for chimney made from briks

$K = 1.1$ for chimney made from steel

4.4.4 Condition for maximum discharge through a chimney

[Dec. '13, Jan.'13, June'12, June'11]

As we know the mass of gases flowing through the chimney

$$\dot{m}_g = \rho_g \cdot A \cdot C$$

Also, velocity of gases $C = \sqrt{2g(H_1 - h')}$ and density of flue gases $\rho_g = \frac{p}{RT_g}$

$$\therefore \dot{m}_g = \frac{p}{RT_g} \cdot A \cdot \sqrt{2g(H_1 - h')}$$

Neglecting pressure loss in the chimney $h' = 0$

$$\therefore \dot{m}_g = A \cdot \sqrt{2gH_1} \cdot \frac{p}{RT_g} \quad ...(4.8)$$

From equation (4.6), $H_1 = H\left[\left(\frac{m_a}{m_a + 1}\right)\frac{T_g}{T_a} - 1\right]$ putting in equation (4.8), we get

$$\dot{m}_g = A \cdot \sqrt{2gH\left[\left(\frac{m_a}{m_a + 1}\right)\frac{T_g}{T_a} - 1\right]} \cdot \frac{p}{RT_g} \quad ...(4.9)$$

The mass flow of flue gases is maximum when $\frac{d\dot{m}_g}{dT_g} = 0$ as T_a, A, p, R, H, and m_a are fixed quantities.

$$\therefore \frac{d}{dT_g}\left\{A\sqrt{2gH\left[\left(\frac{m_a}{m_a + 1}\right)\frac{T_g}{T_a} - 1\right]} \cdot \frac{p}{RT_g}\right\} = 0$$

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$$\therefore \frac{d}{dT_g}\left[\frac{1}{T_g}\sqrt{\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a}-1}\right]=0$$

$$\therefore \sqrt{\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a}-1}\times\frac{d}{dT_g}\left(\frac{1}{T_g}\right)+\frac{1}{T_g}\frac{d}{dT_g}\left(\sqrt{\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a}-1}\right)=0$$

$$\therefore \left[\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a}-1\right]^{\frac{1}{2}}\times\left(-\frac{1}{T_g^{\,2}}\right)+\frac{1}{T_g}\frac{1}{2}\left[\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a}-1\right]^{-\frac{1}{2}}\times\frac{1}{T_a}\times\left(\frac{m_a}{m_a+1}\right)=0$$

$$\therefore \left(\frac{m_a}{m_a+1}\right)\times\frac{1}{2T_aT_g}=\left[\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a}-1\right]\frac{1}{T_g^{\,2}}$$

$$\therefore \left(\frac{m_a}{m_a+1}\right)\times\frac{1}{2T_a}=\left(\frac{m_a}{m_a+1}\right)\frac{1}{T_a}-\frac{1}{T_g}$$

$$\therefore -\left(\frac{m_a}{m_a+1}\right)\frac{1}{2T_a}=-\frac{1}{T_g}$$

$$\therefore \frac{T_g}{T_a}=2\left(\frac{m_a+1}{m_a}\right) \quad \ldots(4.10)$$

Above equation (4.10) is the condition of maximum discharge through the chimney.

4.4.5 Draught at maximum discharge

The condition $\frac{T_g}{T_a}=2\left(\frac{m_a+1}{m_a}\right)$ of maximum discharge putting in equation (4.6) we get

$$(H_1)_{max}=H\left[\left(\frac{m_a}{m_a+1}\right)\times 2\left(\frac{m_a+1}{m_a}\right)-1\right]=H[2-1]=H$$

and $$(h_w)_{max}=353H\left(\frac{1}{T_a}-\frac{1}{2T_a}\right)=\frac{176.5H}{T_a} \text{ mm of water}$$

4.4.6 Efficiency of chimney

The natural draught in chimney is produced by density difference between outside air and flue gases. In order to get sufficient draught at a given height of chimney, certain minimum

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temperature of flue gases is needed to get density difference. This indicates that draught is created at the cost of thermal efficiency of power plant, since a portion of the heat carried away by flue gases to produce the required draught could have been used either in heating the air or feed water.

$$\text{Efficiency of chimney } \eta_{ch} = \frac{\text{Draught produced by chimney in terms of energy}}{\text{Heat of flue gases utilized by chimney to create draught}}$$

Now, draught produced by chimney $(H_1) = H\left[\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a}-1\right]$ **metre** of flue gases column

$$= H \cdot g\left[\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a}-1\right] J$$

Heat of flue gases utilized by chimney to create draught H_1

$$= C_p(T_g - T_g'), J$$

where T_g = temperature of flue gases when chimney used to create draught

T_g' = temperature of flue gases when chimney not used to create draught (artificial draught is used without loosing heat energy in flue gases)

C_p = specific heat of flue gases

$$\therefore \eta_{ch} = \frac{\frac{Hg}{J}\left[\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a}-1\right]}{C_p(T_g - T_g')}, \text{ for SI unit J = 1} \quad ...(4.11)$$

The efficiency of chimney is directly proportional to H. The efficiency of chimney is less than 20%

4.4.7 Advantages and limitations of chimney or natural draught

Advantages :

(1) Chimney draught does not require any external power to produce draught.
(2) Simple in construction, less cost and has long life.
(3) No mechanical parts and hence maintenance cost is negligible.
(4) Chimney keeps flue gases at a high place in the atmosphere which prevents the contamination of atmosphere and maintains the cleanliness.

Limitations :

(1) The draught produced by chimney depends on height of chimney, atmospheric air temperature and temperature of flue gases leaving through the chimney. Hence, in order to get appropriate draught at given height of chimney, the flue gases have to be

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discharged at comparatively high temperatures resulting in the reduction in overall efficiency of plant. About 20% of heat released by the fuel is lost to the flue gases, otherwise it may be used to heat the feed water and air which is supplied to boiler.

(2) The maximum pressure available for producing natural draught by chimney is hardly 10 to 20 mm of water. Hence it is only used for very small boilers.

(3) Draught created by chimney is dependent on atmospheric air and flue gases conditions, hence there is no flexibility to create more draught under peak load conditions because draught available is constant for given height of chimney. The draught can be increased by allowing the flue gases to leave the chimney at higher temperature. This reduces the thermal efficiency of the plant.

(4) Due to low velocity of air, the mixing process of air and fuel is not proper and hence combustion is very poor. This increases the specific fuel combustion of boiler.

(5) In case of modern power plant, the draught produced by chimney is insufficient for high generating capacity and also reduces thermal efficiency. Therefore, now a days chimney is used with artificial draught system in all power plant only to discharge the flue gases high in the atmosphere to maintain the cleanliness of atmospheric air.

Problem 4.1 *Draught produced by chimney is 2 cm of water column. Temperature of flue gas is 300°C and ambient temperature is 33°C. The flue gases formed per kg of fuel burnt is 24 kg. Neglect the losses and take the diameter of chimney as 1.75 m.*

Calculate : (i) Height of chimney in m and (ii) Mass of flue gases flowing through the chimney in kg/min. **[same as Dec. 2013]**

Solution : Given data : $h_w = 2\text{ cm} = 20\text{ mm}$ of water column, $T_g = 300^\circ\text{C} = 573\text{ K}$, $T_a = 33^\circ\text{C} = 306\text{K}$, $(m_a + 1) = 24$ kg/kg of fuel burnt, $D = 1.75$ m, $H = ?$, $m_g = ?$ (kg/min)

(i) Height of chimney

The draught produced by chimney in terms of mm of water column is given by

$$h_w = 353H\left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a + 1}{m_a}\right)\right] \text{ mm of water}$$

$$\therefore 20 = 353H\left[\frac{1}{306} - \frac{1}{573} \times \left(\frac{24}{23}\right)\right] \quad (\because m_a + 1 = 24\text{ kg}, \therefore m_a = 23\text{ kg})$$

$$\therefore H = 39.158\text{ m}$$

(ii) Mass of flue gases

Draught produced by chimney in terms of metre of flue gases column is given by

$$H_1 = H\left[\left(\frac{m_a}{m_a + 1}\right)\frac{T_g}{T_a} - 1\right]$$

Ans

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$$\therefore H_1 = 39.158 \times \left[\left(\frac{23}{24}\right) \times \left(\frac{573}{306}\right) - 1\right]$$

$$\therefore H_1 = 31.112 \text{ m of flue gases column}$$

The velocity of flue gases $C = \sqrt{2gH_1}$ (if no losses)

$$\therefore C = \sqrt{2 \times 9.81 \times 31.112} = 24.71 m/s$$

Density of flue gases

$$\rho_g = \left(\frac{m_a + 1}{m_a}\right) \times \frac{353}{T_g}$$

$$\therefore \rho_g = \left(\frac{24}{23}\right) \times \left(\frac{353}{573}\right) = 0.643 kg/m^3$$

Now, m_g = mass flow rate of flue gases = $\rho_g \cdot A \cdot C$

$$\therefore m_g = 0.643 \times \frac{\pi}{4}(1.75)^2 \times 24.71 = 38.207 kg/s = 2292.41 \text{ kg/min}$$

Problem 4.2 *A boiler is provided with a chimney of 26 m height. The boiler house temperature is 30°C and temperature of flue gases leaving chimney is 300°C. If air supplied to the boiler is 20 kg/kg of fuel. Estimate : (i) Draught in mm of water. (ii) Velocity of gases passing through chimney with 50% loss of draught in chimney.* **[G.U., Dec. 2005]**

Solution : Given data : $H = 26\text{m}$, $T_a = 30^oC = 303 \text{ K}$, $T_g = 300^oC = 573 \text{ K}$, $m_a = 20$ kg/kg of fuel, $h_w = ?$, C = ?, $h' = 50\%$ of total draught $= 0.5H_1$

(i) Draught in mm of water

The draught in mm of water is given by

$$h_w = 353H\left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a + 1}{m_a}\right)\right] \text{ mm of water}$$

$$\therefore h_w = 353 \times 26 \times \left[\frac{1}{303} - \frac{1}{573}\left(\frac{20+1}{20}\right)\right] = \textbf{13.47 mm of water} \quad \textbf{Ans}$$

(ii) Velocity of gases passing through chimney

The draught produced in terms of height of a column of the flue gases in metre is given by

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$$H_1 = H\left[\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a} - 1\right]$$

$$= 26 \times \left[\left(\frac{20}{20+1}\right) \cdot \frac{573}{303} - 1\right] = 20.83 \text{ m of flue gases column}$$

The velocity of flue gases passing through chimney is given by

$$C = \sqrt{2g(H_1 - h')} = \sqrt{2 \times 9.81 \times (20.83 - 0.5 \times 20.83)} = \mathbf{14.29 \text{ m/s}} \quad \textit{Ans}$$

Problem 4.3 *A chimney 30 m high deals with flue gases at 288°C when the outside air temperature is 21°C. Air supplied for combustion is 18 kg per kg of coal burnt. Calculate: (i) the velocity of flue gases in the chimney, if 50% of the draught is lost in friction at the grate and passage, and (ii) the temperature of flue gases under the condition of maximum discharge.* **[G.U., Dec 2001]**

Solution : Given data : H = 30 m, $T_g = 288^oC = 561$ K, $T_a = 21^oC = 294$ K, $m_a = 18$ kg/kg of fuel, C = ?, $h' = 0.5H_1$, T_g = ? (for maximum discharge).

(i) The velocity of flue gases

Draught in terms of m of flue gases

$$H_1 = H\left[\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a} - 1\right] = 30 \times \left[\left(\frac{18}{18+1}\right) \times \frac{561}{294} - 1\right] = 24.23 \text{ m of flue gases.}$$

Velocity of flue gases

$$C = \sqrt{2g(H_1 - h')} = \sqrt{2 \times 9.81 \times (24.23 - 0.5 \times 24.23)} = \mathbf{15.42 \text{ m/s}} \quad \textit{Ans}$$

(ii) The temperature of flue gases for maximum discharge

The condition of maximum discharge is given by

$$\frac{T_g}{T_a} = 2\left(\frac{m_a+1}{m_a}\right)$$

$$\therefore T_g = 2 \times (294) \times \left(\frac{18+1}{18}\right) = 620.67 \text{ K} = \mathbf{347.67^oC} \quad \textit{Ans}$$

Problem 4.4 *Calculate the static draught produced by a chimney of 32 m height when the mean flue gases temperature is such as to cause maximum discharge through chimney. Assume temperature of atmospheric air is 27°C.*

Solution : Given data : h_w = ?, H = 32 m, T_a = 300 K,

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The static draught produced by chimney in terms of mm of water is given by

$$h_w = 353H\left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a + 1}{m_a}\right)\right] \quad ...(i)$$

But, for maximum discharge $\frac{T_g}{T_a} = 2\left(\frac{m_a + 1}{m_a}\right)$

$$\therefore \quad \frac{1}{T_g}\left(\frac{m_a + 1}{m_a}\right) = \frac{1}{2T_a} \quad ...(ii)$$

Equation (ii) is putting in equation (i), we get,

$$\therefore \quad h_w = 353H\left[\frac{1}{T_a} - \frac{1}{2T_a}\right] = \frac{353H}{2T_a} = \frac{176.5H}{T_a}$$

$$\therefore \quad h_w = \frac{176.5 \times 32}{300} = \textbf{18.83 mm of water column} \qquad \textbf{Ans}$$

Problem 4.5 *Determine the height and diameter of chimney to produce a static draught of 18 mm of water column if mean flue gases temperature and flow rate are 300°C and 2100 kg/min respectively. The atmospheric air temperature is 25°C. The gas constant for air is 287 J/kg K and for flue gases is 250 J/kg K. Assume no loss of draught in the chimney and barometer reading as 760 mm of mercury.*

Solution : Given data : $h_w = 18$ mm of water, $T_g = 300^oC = 573$ K, $m_g = 2100$ kg/min $= 35$ kg/s, $T_a = 25^oC = 298$ K, $R_a = 287$ J/kg K, $R_g = 250$ J/kg K, $p_a = 758$ mm of Hg $= 1.01325$ bar, H = ?, D = ?

(i) Height of chimney (H)

$$\text{Density of air } \rho_a = \frac{p_a}{R_a T_a} = \frac{1.01325 \times 10^5}{287 \times 298} = 1.185 \text{ kg/m}^3$$

$$\text{Density of flue gases } \rho_g = \frac{p_a}{R_g T_g} = \frac{1.01325 \times 10^5}{250 \times 573} = 0.707 \text{ kg/m}^3$$

Now, draught produced by chimney

$\Delta p = (\rho_a - \rho_g)gH$, where Δp = in N/m^2

$$\therefore \frac{\rho_w h_w \times g}{1000} = (\rho_a - \rho_g)gH \text{, where } h_w = \text{in mm of water}$$

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$$\therefore h_w = (\rho_a - \rho_g)H$$

$$\therefore 18 = (1.185 - 0.708) \times H$$

$\therefore$ H = height of chimney = **37.65 m**

Ans

(ii) Diameter of chimney

$$\text{Density of flue gases } \rho_g = \left(\frac{m_a + 1}{m_a}\right) \times \frac{353}{T_g}$$

$$\therefore 0.707 = \frac{m_a + 1}{m_a} \times \frac{353}{573}$$

$$\therefore (m_a + 1) = 1.1444 m_a \quad \therefore m_a = 6.77 \text{ kg/kg of fuel}$$

Draught in terms of hot gases column

$$H_1 = H\left[\left(\frac{m_a}{m_a + 1}\right)\frac{T_g}{T_a} - 1\right] = 37.65\left[\left(\frac{6.77}{6.77 + 1}\right) \times \frac{573}{298} - 1\right]$$

$$= 25.426 \text{ m of flue gases column}$$

The velocity of flue gases $C = \sqrt{2gH_1} = \sqrt{2 \times 9.81 \times 25.426} = 22.33$ m/s

Mass flow rate of flue gases is given by

$$\dot{m}_g = \rho_g \cdot A \cdot C = \rho_g \times \frac{\pi}{4} D^2 \times C$$

$$\therefore 35 = 0.707 \times \frac{\pi}{4} D^2 \times 22.33$$

$\therefore D$ = diameter of chimney = **1.68 m** **Ans**

Problem 4.6 *Determine the efficiency of chimney of height 30 m in natural draught system. The temperature of flue gases leaving the chimney is 295°C and boiler surrounding temperature is 18°C. Mass of air supplied to the boiler is 25 kg per kg of fuel. If artificial draught is used for same boiler, the flue gases leaving the boiler would be 120°C. Assume the specific heat of flue gases as 1.2 kJ/kg K.*

Solution : Given data : $H = 30$ m, $T_g = 295°C$ (for natural draught), $T_g' = 120°C$ (for artificial draught), $T_a = 18°C$, $m_a = 25$ kg/kg of fuel burnt, $C_{pg} = 1.2$ kJ/kg K, $\eta_{ch} = ?$

The efficiency of chimney is given by

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$$\eta_{ch} = \frac{Hg\left[\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a}-1\right]}{C_p(T_g - T_g')}$$

$$\therefore \eta_{ch} = \frac{30\times 9.81\times\left[\left(\frac{25}{25+1}\right)\times\left(\frac{295+273}{18+273}\right)-1\right]}{1.2\times10^3(295-120)} = 0.00123 = \mathbf{0.123\%} \quad \textit{Ans}$$

Problem 4.7 *A boiler is equipped with a chimney to get net draught 15 mm of water column. The temperature of flue gases leaving the chimney and surrounding air are 300°C and 27°C respectively. One kg of air and gases occupies a volume of 0.7734 m^3 and 0.7518 m^3 at NTP respectively. Calculate the height of chimney and mass of flue gases formed per kg of fuel burnt.*

Solution : Given data : $h_w = 15$ mm of water column, $T_g = 300°C = 573$ K, $T_a = 27°C =$ 300 K, Volume of air = 0.7734 m^3/kg at NTP, Volume of gases = 0.7518 m^3/kg at NTP,

$H = ?$, $m_g = ?$

Volume of air at 300 K

$$V_a = V_0 \times \frac{T_a}{T_0} = 0.7734 \times \frac{300}{273} = 0.8498\, m^3/kg$$

The density of air at 300 K $\rho_a = \frac{1}{V_a} = 1.1766\, kg/m^3$

Volume of flue gases at 573 K

$$V_g = V_0 \times \frac{T_g}{T_0} = 0.7518 \times \frac{573}{273} = 1.5779\ m^3/kg$$

The density of flue gases at 573 K, $\rho_g = \frac{1}{V_g} = 0.6337\ kg/m^3$

As we know draught produced by chimney in terms of mm of water column

$$h_w = (\rho_a - \rho_g)H$$

$$\therefore 15 = (1.1766 - 0.6337)H$$

$$\therefore H = \mathbf{27.63\ m} \quad \textit{Ans}$$

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Also, $h_w = 353H\left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right)\right]$

$$\therefore\ 15 = 353\times 27.63\times\left[\frac{1}{300} - \frac{1}{573}\left(\frac{m_a+1}{m_a}\right)\right]$$

$\therefore\ m_a = 34.72$ kg/kg of fuel burnt

Mass of flue gases formed/kg of fuel burnt

$m_g = m_a + 1 = 34.71 + 1 = \mathbf{35.72\ kg}$ ***Ans***

- ***Problem 4.8 :*** *A boiler uses 1000 kg. of coal/h. The temperature of hot gases inside the chimney is 650 K and the outside air temperature is 300 K. The draught produced by the chimney of 25 m height is 15 mm of water. Determine the air supplied per kg. of fuel burnt, draught in terms of hot gases, mass flow rate of hot gases and area of chimney required if the co-efficient of velocity is 0.4.* ***[June '11]***

Solution : Given data : $m_f = 1000\,kg/hr = 0.278\,kg/s$, $T_g = 650\,K$, $T_a = 300\,K$, $H = 25\,m$, $h_w = 15\,mm$ of water.

The draught produced by chimney in terms of mm of water column is given by

$$h_w = 353H\left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right)\right]$$

$$\therefore 15 = 353\times 25\times\left[\frac{1}{300} - \frac{1}{650}\left(\frac{m_a+1}{m_a}\right)\right]$$

$$\therefore \frac{m_a+1}{m_a} = 1.0618$$

$\therefore m_a + 1 = 1.0618\,m_a$

$\therefore\ m_a = \mathbf{16.168\ kg/kg\ of\ fuel\ burnt}$ ***Ans***

$= 16.168\times 1000 = 16168\ kg/hr = \frac{16168}{3600}\ kg/s = 4.49\ kg/s$

Draught produced in terms of m of flue gases is given by

$$H_1 = H\left[\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a} - 1\right]$$

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$$= 25 \times \left[\left(\frac{16.168}{16.168+1} \right) \times \frac{650}{300} - 1 \right]$$

= 26.01 m of flue gases column *Ans*

Mass of flue gases per kg of flue burnt

$$m_g = m_a + 1 = 16.168 + 1 = \textbf{17.168 kg/kg of fuel burnt} \quad \textit{Ans}$$

$$= 17.168 \times \frac{1000}{3600} kg/s = 4.768\ kg/s$$

The velocity of flue gases passing through chimney is given by

$$C = C_v \sqrt{2gH_1} \text{ where } C_v = \text{coefficient of velocity}$$

$$= 0.4\sqrt{2 \times 9.81 \times 26.01} = 9.036\, m/s$$

Now, $m_g = \rho_g\ A \cdot C, kg/s$ and $\rho_g = \left(\frac{m_a + 1}{m_a} \right) \times \frac{353}{T_g}$

$$\therefore m_g = \left(\frac{m_a + 1}{m_a} \right) \times \frac{353}{T_g} \times A \times C$$

$$\therefore 17.168 \times \frac{1000}{3600} = \left(\frac{16.168+1}{16.168} \right) \times \frac{353}{650} \times A \times 9.036$$

$$\therefore A = \textbf{0.91}\ m^2 \quad \textit{Ans}$$

Problem 4.9 : *Determine height of the chimney to produce a static draught of 17 mm of water. The temperature in the chimney is 300°C and surrounding temperature is 28°C. The barometer reads 755 mm of Hg. The value of characteristic gas constant is 287 J/kg K and that for gases is 255 J/kg K.* **[Dec. '11]**

Solution : *Given data :*

h_w = 17 mm of water, T_g = 300°C = 573 K, T_a = 28°C = 301 K, p_a = 755 mm of

$$H_g = \frac{755}{760} \times 101.3 = 100.63\, kN/m^2,\ R_a = 287\ J/kg\ K,\ R_g = 255\ J/kg/K.$$

$$\text{Density of air } \rho_a = \frac{p_a}{R_a T_a} = \frac{100.63 \times 10^3}{287 \times 301} = 1.165\, kg/m^3$$

$$\text{Density of flue gases } \rho_g = \frac{p_a}{R_g T_g} = \frac{100.63 \times 10^3}{255 \times 573} = 0.688\ kg/m^3$$

Now, draught produced by chimney

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$$\Delta p = \left(\rho_a - \rho_g\right) gH, \text{ where } \Delta p = in\ N/m^2$$

$$\therefore \frac{\rho_w g \cdot h_2}{1000} = \left(\rho_a - \rho_g\right) g \cdot H, \text{ where } h_w = \text{in mm of water}$$

$$\therefore h_w = \left(\rho_a - \rho_g\right) H$$

$$\therefore \text{Height of chimney } H = \frac{h_w}{\left(\rho_a - \rho_g\right)} = \frac{17}{(1.165 - 0.688)} = \mathbf{35.64\,m} \quad \textbf{Ans}$$

Problem 4.10 : *A 40 m high chimney is discharging flue gases at 350°C, when the ambient temperature is 30°C. The quantity of air supplied is 18 kg/kg of fuel burnt. Determine:*

(i) Draught produced in mm of water.

(ii) Equivalent draught in m of hot gas column.

(iii) Efficiency of chimney if minimum temperature of artificial draught is 150°C mean specific heat of flue gases is 1.005 kJ/kgK,

(iv) Percentage of heat spent in natural draught system if calorific value of fuel supplied is 30,600 kJ/kg.

(v) Temperature of chimney gases for maximum discharge in given time and what will be corresponding draught in mm of water produced. ***[June '12]***

Solution : *Given data : H = 40 m,* $T_g = 350°C = 623\ K$, $T_a = 30°C = 303\ K$, $m_a = 18$ kg/kg of fuel burnt, $h_w = ?$, $H_1 = ?$, $\eta_{ch} = ?$ (for $T_g' = 150°C$), $C_{pg} = 1.005$ kJ/kgK, % of heat spent = ?, T_g (for maximum discharges) = ?

(i) Draught produced in mm of water

$$h_w = 353H\left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a + 1}{m_a}\right)\right]$$

$$= 353 \times 40 \times \left[\frac{1}{303} - \frac{1}{623}\left(\frac{18+1}{18}\right)\right]$$

$$= \mathbf{70.52\ mm\ of\ water}$$

(ii) Equivalent draught in m of hot gases column **Ans**

$$H_1 = H\left[\left(\frac{m_a}{m_a + 1}\right)\frac{T_g}{T_a} - 1\right] = 40 \times \left[\left(\frac{18}{18+1}\right) \times \frac{623}{303} - 1\right]$$

$$= \mathbf{37.91\ m\ of\ flue\ gases\ column}$$

Ans

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(iii) Efficiency of chimney

$$\eta_{ch} = \frac{H \cdot g \cdot \left[\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a} - 1\right]}{C_{pg}\left(T_g - T_g'\right)}$$

$$= \frac{H_1 \times g}{C_{pg}\left(T_g - T_g'\right)} = \frac{37.91 \times 9.8}{1.005 \times 10^3 (350 - 150)} \times 100$$

$$= \mathbf{0.18\ \%} \quad \textit{Ans}$$

(iv) Percentae of heat spent in natural draught system

$$= C_{pg}\left(T_g - T_g'\right)$$

$$= 1.005\,(350 - 150)$$

$$= \mathbf{201\ kJ} \quad \textit{Ans}$$

(v) Temperature of chimney gases for maximum discharge

Condition of maximum discharge is given by

$$\frac{T_g}{T_a} = 2\left(\frac{m_a+1}{m_a}\right)$$

$$\therefore T_g = 2 \times 303 \times \left(\frac{18+1}{18}\right) = 639.67\,K = \mathbf{366.67^oC} \quad \textit{Ans}$$

4.5 Mechanical draught

The draught required in high capacity modern steam power plant may vary form 30 to 350 mm of water in order to maintain flow of large volume of air and gases (4×10^5 to 8×10^5 m^3/min). Also, large steam power plant is equipped with economizer and air preheater, the exit temperature of flue gases is sufficiently reduced. In such case in order to produce certain amount of draught the height of chimney may be high enough. It has been seen that draught produced by chimney hardly 10 to 20 mm of water. Also, it has effected by atmospheric conditions, no flexibility, poor efficiency and loss of energy. It is not possible to produce a draught more than 40 mm water with the help of natural draught system (chimney) because it is neither economical nor convenient to build a chimney of enough height for larger amount of draught. Therefore it becomes necessary to provide mechanical means to meet the larger draught requirements. This mechanical arrangement used to produce larger draught is called ***artificial or mechanical draught*** which independent on climate conditions and flexible to meet fluctuating load on boiler. Boiler with mechanical draught do not need so high a chimney

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as is necessary with natural draught. Chimney is needed in mechanical draught some times to avoid the smoke, dust, cinder and ash from falling over the adjacent buildings. The mechanical draught is created by fan. There are three types of mechanical draughts as Forced draught, Induced draught and Balanced draught.

4.5.1 Forced draught

In a forced draught system, the pressure over boiler settings is normally maintained above atmospheric pressure. Therefore this draught is also known as positive draught. In this draught, a blower/fan is installed near the base of the boiler as shown in Fig. 4.2 and air is forced to pass through the furnace, flues, economizer, air-preheater and to the stack or chimney. A stack/chimney in this draught system required to discharge gases high in the atmosphere to prevent the contamination. It is not much significant for producing draught therefore height of the chimney may not be very high.

Forced draught is commonly used with under feed stokers, as considerable pressure is needed to force the air required for combustion up through the deep fuel bed.

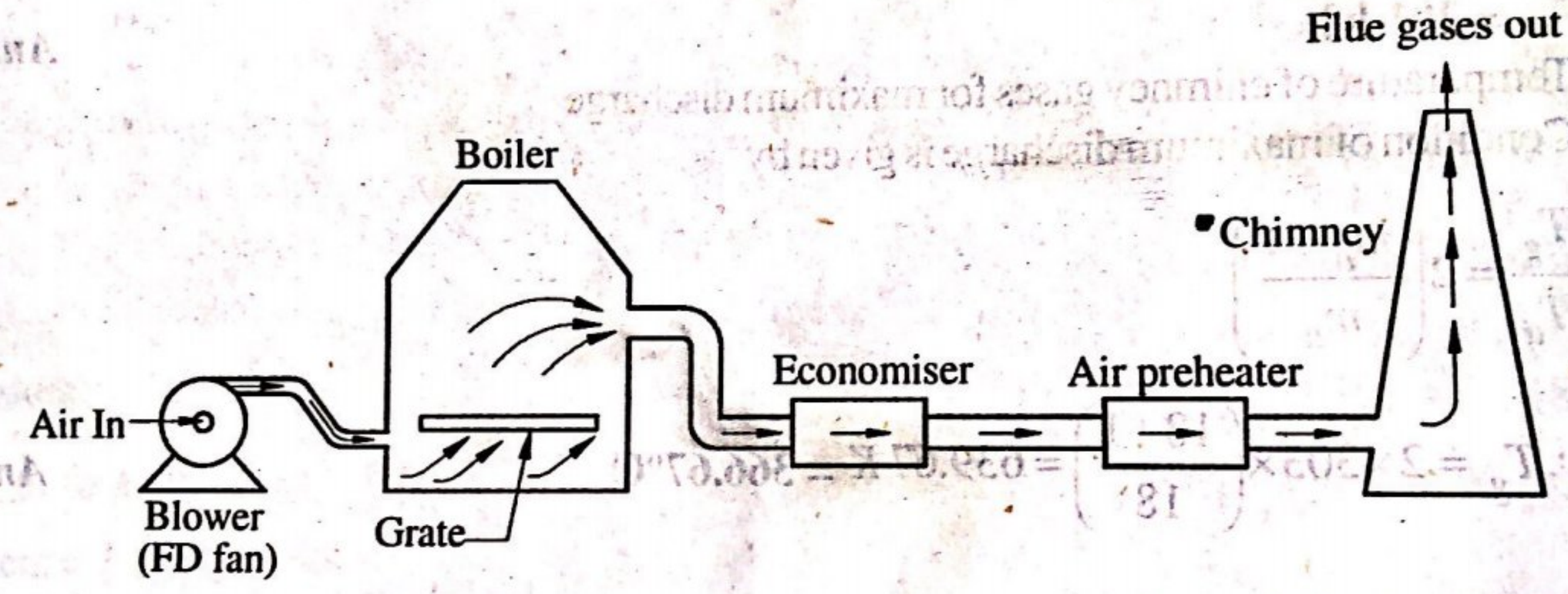


Fig. 4.2 Forced draught

4.5.2 Induced draught

In an induced draught system, the pressure over the fuel bed is maintained below the atmosphere. The blower or fan is installed near or at base of the chimney as shown in Fig. 4.3 and sucks the burned gases from the furnace. Hence pressure inside the furnace is reduced below atmosphere and induces the atmospheric air to flow through the furnace. The action of the induced draught is similar to the action of the chimney. But draught produced by induced draught system is independent of the temperature of the hot gases therefore the flue gases may be discharged as cold as possible in economiser and air preheater.

Induced draught is used generally when economizer and air preheater are incorporated in the system. The induced draught fan or blower should be placed such that the temperature of gas handled is lowest possible. This clearly suggests that it should be installed after the air preheater. In an induced draught system, fan deals with hot flue gases hence it has to handles

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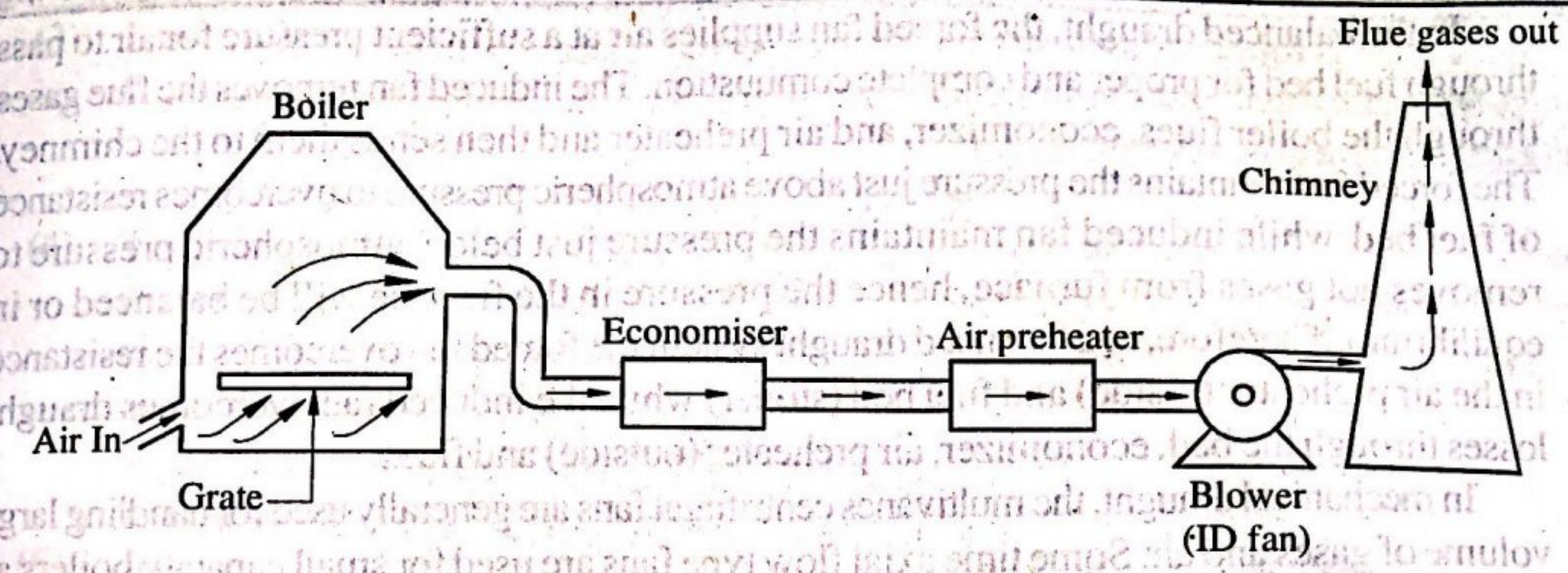


Fig. 4.3

large volume of hot gases. Therefore, size of induced draught fan and power required are more compared to forced draught fan.

4.5.3 Balanced draught

The balanced draught is a combination of forced, induced draught and chimney. In this system the forced fan is located at base of boiler and induced fan is located at base of chimney as shown in Fig. 4.4. Air is push by forced fan and flue gases is sucked by induced fan.

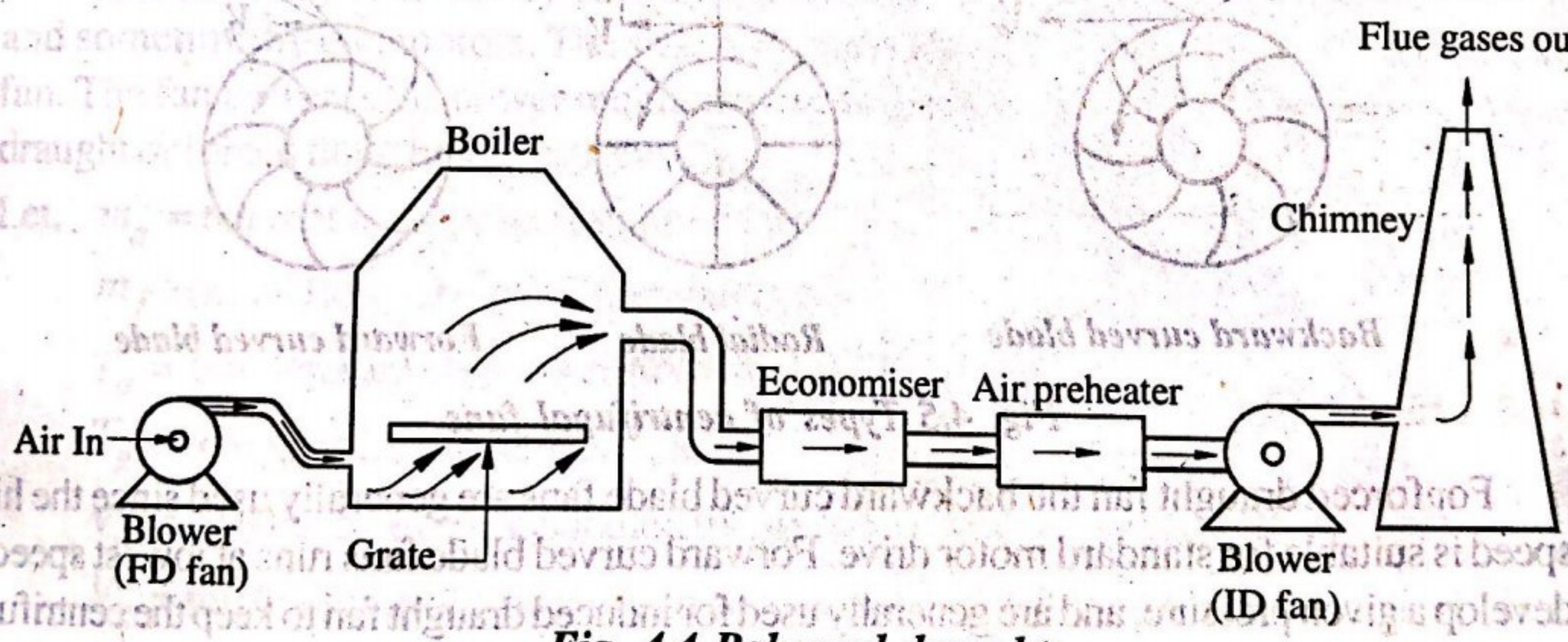


Fig. 4.4 Balanced draught

As we know that the forced draught system the pressure inside furnace is above atmosphere. It forced draught is used alone, then furnace can not be opened either for firing or inspection because the high pressure air inside the furnace will try to blow out suddenly. Therefore, there is chance of blowing out the fire completely and furnace stops. Similarly if the induced draught is used alone, then also furnace can not be opened either for firing or inspection because the cold air will try to rich into the furnace as the pressure inside the furnace is below the atmospheric pressure. This reduces the effectiveness of draught and dilutes the combustion. The balanced draught overcomes both the difficulties as above either using forced draught or induced draught alone.

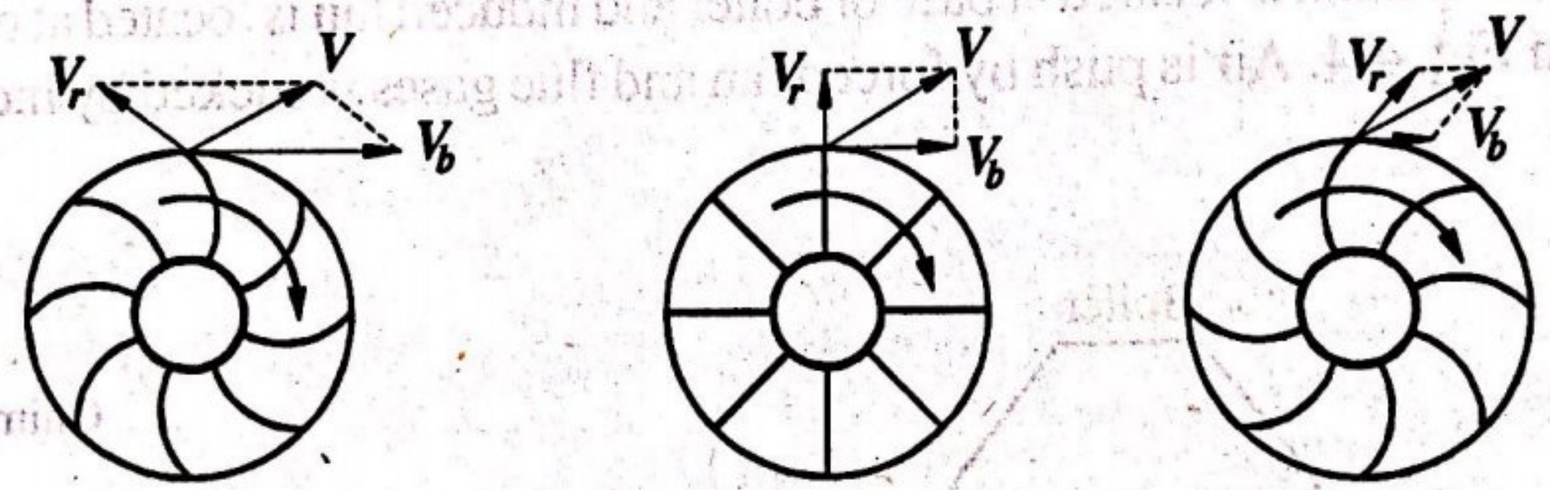
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In the balanced draught, the forced fan supplies air at a sufficient pressure for air to pass through fuel bed for proper and complete combustion. The induced fan removes the flue gases through the boiler flues, economizer, and air preheater and then sends them to the chimney. The forced fan maintains the pressure just above atmospheric pressure to overcomes resistance of fuel bed while induced fan maintains the pressure just below atmospheric pressure to removes hot gases from furnace, hence the pressure in the furnace will be balanced or in equilibrium. Therefore, in a balanced draught system the forced fan overcomes the resistance in the air preheater (inside) and fuel bed (stoker) while the induced fan overcomes draught losses through the bed, economizer, air preheater (outside) and flues.

In mechanical draught, the multivanes centrifugal fans are generally used for handling large volume of gases and air. Some time axial flow type fans are used for small capacity boilers as they develop only small static pressure. In the centrifugal flow fans the air is compressed by means of a rotor rotating in spiral shape casing.

The performance of a centrifugal type fan depends on the shapes of blades which are three types (Refer Fig. 4.5) as (i) Backward curved blades, (ii) Forward curved blades and (iii) Straight radial blades. The backward curved blades gives a higher speed than the straight radial blades and radial blades gives a higher speed than forward curved blades.



Backward curved blade *Radial blade* *Forward curved blade*

Fig. 4.5 Types of centrifugal fans

For forced draught fan the backward curved blade fans are generally used since the high speed is suitable for standard motor drive. Forward curved blade fans runs at lowest speed to develop a given pressure, and are generally used for induced draught fan to keep the centrifugal stresses small. The straight radial blades also recommended for induced draught fans.

4.5.4 Comparision between forced and induced draught

Forced draught	Induced draught
(1) Fan is installed near or at the base of the boiler	(1) Fan is installed at base of chimney
(2) It handles low temperature and less volume of air, hence size and power required for forced fan is less than induced fan	(2) It handles more volume of gases at higher temperature. Therefore, size and power required for induced fan is more than forced fan.

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(3) Fan handles air and thus water cooled bearing are not required	(3) Fan handles flue gases of high temperature and thus water cooled bearings are needed for induced draught fan.
(4) The pressure inside the furnace is above the atmospheric pressure, hence there is no chance of air leakage. However, when doors are opened for firing or inspection the high pressure air inside the furnace will try to blow out suddenly.	(4) The pressure inside the furnace is below the atmospheric pressure, hence there is continuous air leakage is possible.
(5) There is uniform flow of air through the grate and furnace	(5) Flow of air is not uniform
(6) The heat transfer efficiency will be higher than induced draught	(6) The heat transfer efficiency will be lower than forced draught.
(7) Lower investment and operating costs compared to induced draught.	(7) Higher investment and operating cost compared to forced draught.

4.5.5 Power requirement by fans

The fans may be driven by variable speed motors, constant speed motors, steam turbine and sometime by d.c. motors. The size of motor is chosen based on power required to the fan. The fan power is the power required to move gases or air by the fan in case of induced draught or forced draught fan respectively.

Let, m_a = mass of air supplied per kg of fuel

m_f = mass flow rate of fuel supplied, kg/s

T_a = temperature of boiler surrounding air, K

T_g = temperature of flue gases, K

V_a = volume of air to be handled by the fan, m^3/s

V_g = total volume of flue gases to be handled by the fan, m^3/s

η_{fm} = mechanical efficiency of fan

(1) Power required for forced draught fan (P_{FD})

Mass of air supplied in kg/sec = m_a (in kg/kg of fuel) $\times m_f$ (kg/sec)

Volume of air is handled by FD fan

$$V_a = \frac{\text{Mass of air supplied (kg/sec)}}{\text{Density of air}} = \frac{m_a \cdot m_f}{\rho_a}$$

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But $\rho_a = \frac{353}{T_a}$

$$\therefore V_a = \frac{m_a \cdot m_f}{353} \cdot T_a, \ m^3/s$$

P_{FD} = Draught created by fan (in N/m^2) × Volume of air (in m^3/s)

$= \Delta p \times V_a$

$$= \rho_w g \cdot h_w \times V_a = 1000 \times g \times \frac{h_w}{1000} \times \frac{m_a m_f T_a}{353}, \text{ Nm/s}$$

(where h_w in *mm* of water)

$$= \frac{9.81 \cdot h_w m_a \cdot m_f \cdot T_a}{353 \times 1000}, \text{ kW}$$

$$\therefore P_{FD} = 2.779 \times 10^{-5} h_w \cdot m_a \cdot m_f \cdot T_a, \ kW \quad ...(4.12)$$

Power required for FD fan motor $P = \frac{P_{FD}}{\eta_{fm}}$

(2) Power required for induced draught fan (P_{ID})

Mass of flue gases in kg/s = $(m_a + 1) \cdot m_f$

Volume of flue gases is handled by ID fan

$$V_g = \frac{\text{Mass of flue gases (kg/s)}}{\text{Density of flue gases}} = \frac{(m_a + 1) m_f}{\rho_g}$$

But $\rho_g = \left(\frac{m_a + 1}{m_a}\right) \times \frac{353}{T_g}$

$$\therefore V_g = \frac{(m_a + 1) m_f}{\left(\frac{m_a + 1}{m_a}\right) \times \frac{353}{T_g}} = \frac{m_a \cdot m_f}{353} \cdot T_g$$

P_{ID} = Draught created by fan (in N/m^2) × Volume of flue gases (in m^3/s)

$= \Delta p \times V_g$

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$$\therefore P_{ID} = \rho_w g h_w \times V_g = \frac{9.81 h_w m_a m_f T_g}{353 \times 1000}, kW$$

$$\therefore P_{ID} = 2.779 \times 10^{-5} h_w \cdot m_a \cdot m_f \cdot T_g, \text{kW} \quad ...(4.13)$$

Power required for ID fan motor $P = \frac{P_{ID}}{\eta_{fm}}$

The ratio of P_{ID} and P_{FD} for same draught

$$\frac{\text{Power of ID fan}}{\text{Power of FD fan}} = \frac{T_g}{T_a}$$

Problem 4.11 *Calculate required motor capacity needed for I.D. and F.D. for following condition.*

Air supplied per kg of fuel in each case = 20 kg.
Mass of coal burnt per hour = 1980 kg.
Draught required = 50 mm of water.
Temperature of flue gases leaving the boiler in each case = 250°C.
Temperature of air surrounding the boiler = 21°C.
Efficiency of fans = 80%.

Solution : Given data : $m_a = 20$ kg/kg of fuel, $m_f = 1980$ kg/hr $= \frac{1980}{3600} = 0.55$ kg/s, h_w = 50 mm of water, $T_g = 250°C = 523$ K, $T_a = 21°C = 294K$, $P_{FD} = ?$, $P_{ID} = ?$

Power required to drive the forced draught fan

$$P_{FD} = 2.779 \times 10^{-5} \cdot h_w \cdot m_a \cdot m_f \cdot T_a$$

$$= 2.779 \times 10^{-5} \times 50 \times 20 \times 0.55 \times 294$$
$$= 4.49 \text{ kW}$$

Motor capacity for FD fan $= \frac{4.49}{0.8} = \mathbf{5.61\ kW}$ ***Ans***

Power required to drive the Induced draught fan

$$P_{ID} = 2.779 \times 10^{-5} \cdot h_w \cdot m_a \cdot m_f \cdot T_g$$

$$= 2.779 \times 10^{-5} \times 50 \times 20 \times 0.55 \times 523$$
$$= 7.99 \text{ kW}$$

Motor capacity for I.D. fan $= \frac{7.99}{0.8} = \mathbf{9.99\ kW}$ ***Ans***

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Problem 4.12 *Compute the required motor capacity needed for F.D. fan under following condition :*

Coal rate : 10 tonnes/hr
Coal analysis : C78.0, H3.0, O3.0, S1.0, M7.0, A8.0
Excess air : 30%
Plenum chamber pressure : 180 mm of water gauge
Mechanical efficiency of the fan : 60%
Room temperature : 30°C **[Dec 2010, 07 marks]**

Solution : Given data : $m_f = 10$ tonnes/hr $= 2.7778$ kg/s, $h_w = 180$ mm of water,

$\eta_{fm} = 0.6$, $T_a = 30^{\circ}C = 303K$,

The stoichiometric air required

$$= \frac{100}{23}\left[\frac{8}{3}C + 8\left(H - \frac{O}{8}\right) + S\right] \text{ kg/kg of fuel burnt}$$

$$= \frac{100}{23}\left[\frac{8}{3}(0.78) + 8\left(0.03 - \frac{0.03}{8}\right) + 0.01\right]$$

$= 10$ kg/kg of fuel burnt

Actual air supplied = Stoichiometric air + Excess air

$$m_a = 10 + \frac{30}{100} \times 10 = 13 \text{ kg/kg of fuel}$$

Power required to drive the F.D. fan

$$P_{FD} = 2.779 \times 10^{-5} \cdot h_w \cdot m_a \cdot m_f \cdot T_a$$

$$= 2.779 \times 10^{-5} \times 180 \times 13 \times 2.7778 \times 303$$

$$= 54.73 \text{ kW}$$

$$\text{Motor capacity} = \frac{P_{FD}}{\eta_{fm}} = \frac{54.73}{0.6} = \mathbf{91.22\ kW}$$ ***Ans***

Problem 4.13 *Calculate motor power required for I.D. fan under following condition :*

Air supplied per kg of fuel burnt = 25 kg
Flow rate of fuel = 0.6 kg/s,
Draught created in mm of water = 100,
Leakages of air in draught system = 20%,
Mean temperature of flue gases = 200°C,
Temperature of boiler surrounding air = 25°C
Mechanical efficiency of ID fan = 65%

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Solution : Given data : $m_a = 25$ kg/kg of fuel, $m_f = 0.6$ kg/s, $h_w = 100$ mm of water,

Leakages of air = 20%, $T_g = 200°C = 473$ K, $T_a = 25°C = 298$ K, $\eta_{fm} = 0.65$, $P_{ID} = ?$

Volume of flue gases to be handled by I.D. fan

$$V_g' = \frac{m_a : m_f}{353} \cdot T_g = \frac{25 \times 0.6}{353} \times 473 = 20.1 m^3/\text{sec}$$

Actual volume of flue gases

$$V_g = V_g' + \text{Leakages}$$

$$= 20.1 + \frac{20}{100} \times 20.1 = 24.12 m^3/s$$

Power required to drive I.D. fan

$$P_{ID} = \rho_w g h_w \times V_g$$

$$= g h_w \times V_g \quad \text{where } h_w \text{ in mm of water}$$

$$= 9.81 \times 100 \times 24.12 = 23660.7 \text{ W} = 23.66 \text{ kW}$$

$$\text{Motor power} = \frac{23.66}{0.65} = \mathbf{36.4\ kW}$$ ***Ans***

Problem 4.14 *Draught loss through fuel bed, boiler, bends, and chimney are 10 mm, 15 mm, 4 mm, and 3 mm of water respectively. The velocity of flue gases leaving the chimney is 8 m/s. The actual draught created is 80% of theoretical draught. The flue gases mean temperature is 280°C. The mass of air required is 15 kg/kg of fuel. The surrounding air temperature is 27°C. Calculate power required to drive the I.D. fan. Assume fuel flow rate 1 kg/s.*

Solution : Given data : $m_a = 15$ kg/kg of fuel, $m_f = 1$ kg/s, $T_g = 280°C = 553$ K,

$v_g = 8$ *m/s*, $T_a = 27°C = 300$K, $P_{ID} = ?$

$$\text{Density of flue gases } \rho_g = \left(\frac{m_a + 1}{m_a}\right) \times \frac{353}{T_g} = \left(\frac{15+1}{15}\right) \times \frac{353}{553} = 0.681 kg/m^3$$

Pressure head equivalent to velocity of flue gases at exit

$$h_g = \frac{v_g}{2g} = \frac{(8)^2}{2 \times 9.81} = 3.262 \text{ m of flue gases} = 3.262 \times \frac{\rho_g}{\rho_w} \text{ m of water}$$

$$\left(\because \rho_g g h_g = \rho_w g h_w\right)$$

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$$= 3.262 \times \frac{0.681}{1000} = 2.22 \text{ mm of water}$$

Total draught required h_w = draught losses

$$\therefore\ h_w = 10 + 15 + 4 + 3 + 2.22 = 34.22 \text{ mm of water}$$

Power required to drive I.D. fan

$$P_{ID} = 2.779 \times 10^{-5} \cdot h_w \cdot m_a \cdot m_f \cdot T_g$$

$$= 2.779 \times 10^{-5} \times 34.22 \times 15 \times 1 \times 553$$

$$= 7.888 \text{ kW} \quad \textit{Ans}$$

Problem 4.15 *Forced draught fan discharges 1500 m^3 of air per minute through the outlet of 1.7 m diameter and maintains a static pressure of 110 mm of water. The temperature of air is 25°C. The density of air at NTP is 1.293 kg/m^3. Calculate power of motor to drive the F.D. fan if efficiency of fan is 70%.*

Solution : $V_a = 1500 \text{ m}^3/\text{min} = 25 \text{ m}^3/\text{s}$, Diameter of outlet of fan $D_e = 1.7$ m, $h_w = 110$ mm of water, $T_a = 25°\text{C} = 298$ K, ρ_a (at NTP) $= 1.293 \text{ kg/m}^3$, $\eta_{fm} = 0.7$, $P_{ID} = ?$

Density of air at 273 K = 1.293 kg/m³

$$\therefore \text{ Density of air at 298 K} = \frac{1.293 \times 273}{298} = 1.184\, kg/m^3$$

$$\text{Velocity of air at outlet of fan} = \frac{V_a}{\frac{\pi}{4}(D_e)^2} = \frac{25}{\frac{\pi}{4}(1.7)^2} = 11.01\, m/s$$

$$\text{Velocity head} = \frac{v^2}{2g} = \frac{(11.01)^2}{2 \times 9.81} = 6.18\, m$$

$$\text{Velocity head in terms of mm of water} = \frac{1.184}{1000} \times 6.18 \times 1000 = 7.317 \text{ mm of water}$$

Total draught created by fan = Static draught + Velocity head

$$= 110 + 7.317 = 117.32 \text{ mm of water}$$

Power required to drive the F.D. fan

$$P_{FD} = \rho_w g h_w \times V_a = 1000 \times 9.81 \times \frac{h_w}{1000} \times V_a$$

$$= 9.81 \times 117.32 \times 25 = 28772 W = 28.77 \text{ kW} \quad \textit{Ans}$$

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Problem 4.16 *A draught of 80 mm of water is produced by using I.D. fan. The temperature of the flue gases and ambient air are 200°C and 27°C respectively. The fuel supplied rate is 0.6 kg/sec and air fuel ratio used is 15:1. The efficiency of fan is 75%. Calculate the power of prime mover required to run the fan. If the I.D. fan is replaced by F.D. fan assuming other data same, calculate the percentage reduction in power required to run fan.*

Solution : Given data : $h_w = 80$ mm, $T_g = 200°C = 473$ K, $T_a = 27°C = 300$ K, $m_f = 0.6$ kg/s, $m_a / m_f = 15$, $\eta_{fm} = 0.75$, $P_{ID} = ?$

Power required to run I.D. fan

$$P_{ID} = 2.779 \times 10^{-5} \times h_w \cdot m_a \cdot m_f \cdot T_g$$

$$= 2.779 \times 10^{-5} \times 80 \times 15 \times 0.6 \times 473$$

$$= 9.464 \text{ kW}$$

$$\text{Motor power} = \frac{9.464}{0.75} = \mathbf{12.618\ kW}$$ ***Ans***

Power required to run F.D. fan

$$P_{FD} = P_{ID} \times \frac{T_a}{T_g} = 9.464 \times \frac{300}{473} = 6\,kW$$

$$\text{Motor power} = \frac{6}{0.75} = 8\,kW$$

Percentage reduction in power

$$= \frac{12.618 - 8}{12.618} \times 100 = \mathbf{36.6\ \%}$$ ***Ans***

Problem 4.17 *1500 m^3 of flue gases per minute are discharged by an induced draught fan through a chimney of diameter of 1.8 m at 120°C. The total losses in the system are equivalent to 120 mm of water column. The efficiency of fan is 80%. Calculate power required by motor to drive the fan. Assume R = 255 kJ/kg K for gases and atmospheric pressure is 1 bar.*

Solution : Given data : $V_g = 1500$ m^3/min = 25 m^3/s, D = 1.8 m, $T_g = 120°C = 393$ K, $\eta_{fm} = 0.8$, $P_{ID} = ?$, R = 255 kJ/kg K (for gases), $p_a = 1$ bar.

Density of flue gases at atmospheric pressure

$$\rho_g = \frac{p_a}{RT_g} = \frac{1 \times 10^5}{255 \times 393} = 0.997\,kg/m^3$$

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Velocity of flue gases at exit

$$v_g = \frac{V_g}{\frac{\pi}{4}D^2} = \frac{25}{\frac{\pi}{4}(1.8)^2} = 9.824\, m/s$$

Pressure head equivalent to velocity of flue gases at exit

$$= \frac{v_g^{\,2}}{2g} = \frac{(9.824)^2}{2\times 9.81} = 4.92 \text{ m of flue gases}$$

$$= 4.92 \times \frac{\rho_g}{\rho_w} \times 1000 \text{ mm of water}$$

$$= 4.92 \times \frac{0.997}{1000} \times 1000 = 4.9 \text{ mm of water}$$

Total draught required

h_w = Total losses in system + Velocity head at exit

= 120 + 4.9 = 124.9 mm of water

Power required to drive I.D. fan $= g \cdot h_w \times V_g = 9.81 \times 124.9 \times 25 = 30.63\, kW$

Motor power $= \frac{30.63}{0.8} = \mathbf{38.29\ kW}$ ***Ans***

4.6 Advantages and disadvantages of mechanical draught over natural draught [May '13]

Mechanical draught	Natural draught
Advantages :	**Disadvantages :**
(1) It does not depend on the atmospheric temperature.	(1) It depends on atmospheric temperature.
(2) Chimney height is reduced since the function of chimney is only to discharge flue gases at a convenient height in the atmosphere.	(2) In order to create required draught the chimney height is more.
(3) It improves the rate of combustion as draught is more. The better distribution and mixing of air with fuel is possible hence quantity of air required per kg of fuel is less. This reduces mass of flue gases formed and heat carried by exhaust gases.	(3) Poor combustion due to less draught. Hence more quantity of air required. This increases mass of flue gases formed and heat carried by exhaust gases.

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(4) Better control of combustion and evaporation rate since the air flow can be regulated as per requirement for combustion.
(5) The heat of flue gases can be recovered using economizer, air preheater since the flue gases temperature can be reduced. Hence thermal efficiency of plant is high.
(6) Less fuel consumption for given power
(7) Low grade fuel can be used due to high intensity of draught and combustion.
(8) Efficiency of draught about 7%.

Disadvantages :

(1) High investment cost.
(2) External power is required to drive the fans.
(3) High operating and maintenance costs.

(4) Combustion rate can not be controlled since air flow can not be regulated.
(5) The heat of flue gases can not be recovered since the flue gases have to be discharged at high temperature in order to create appropriate draught Hence thermal efficiency of plant is low.
(6) More fuel consumption for given power.
(7) High grade fuel are used.
(8) Efficiency of draught hardly about 1%

Advantages :

(1) Less investment cost.
(2) No external power is required.
(3) Operating and maintenance costs is practically nil.

4.7 Steam jet draught

The artificial draught created by the jet of steam is called steam jet draught. The steam jet draught is a simple and easy method of producing artificial draught. It may also be of the forced draught type or induced draught type.

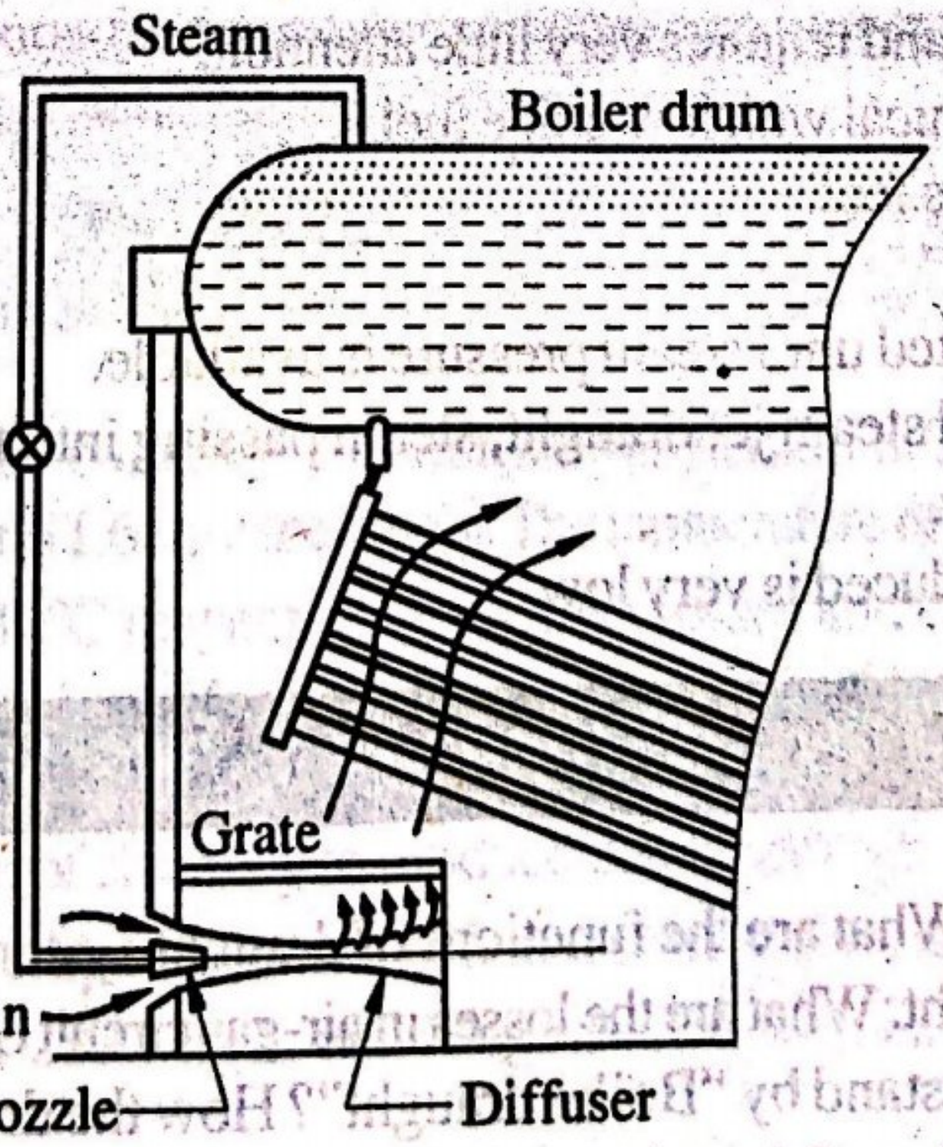


Fig. 4.6 Forced steam jet draught

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In forced steam jet draught as shown in Fig. 4.6 steam (from boiler drum) is supplied to the nozzle through throttle valve. The nozzle is placed into a diffuser pipe which located below the grate of boiler. The steam comes out of the nozzle with high velocity and draws air along with it. The kinetic energy of the mixture (steam and air) is converted into pressure energy when it passes through the diffuser pipe. Therefore air is forced through the fuel bed, furnace and passed through the chimney.

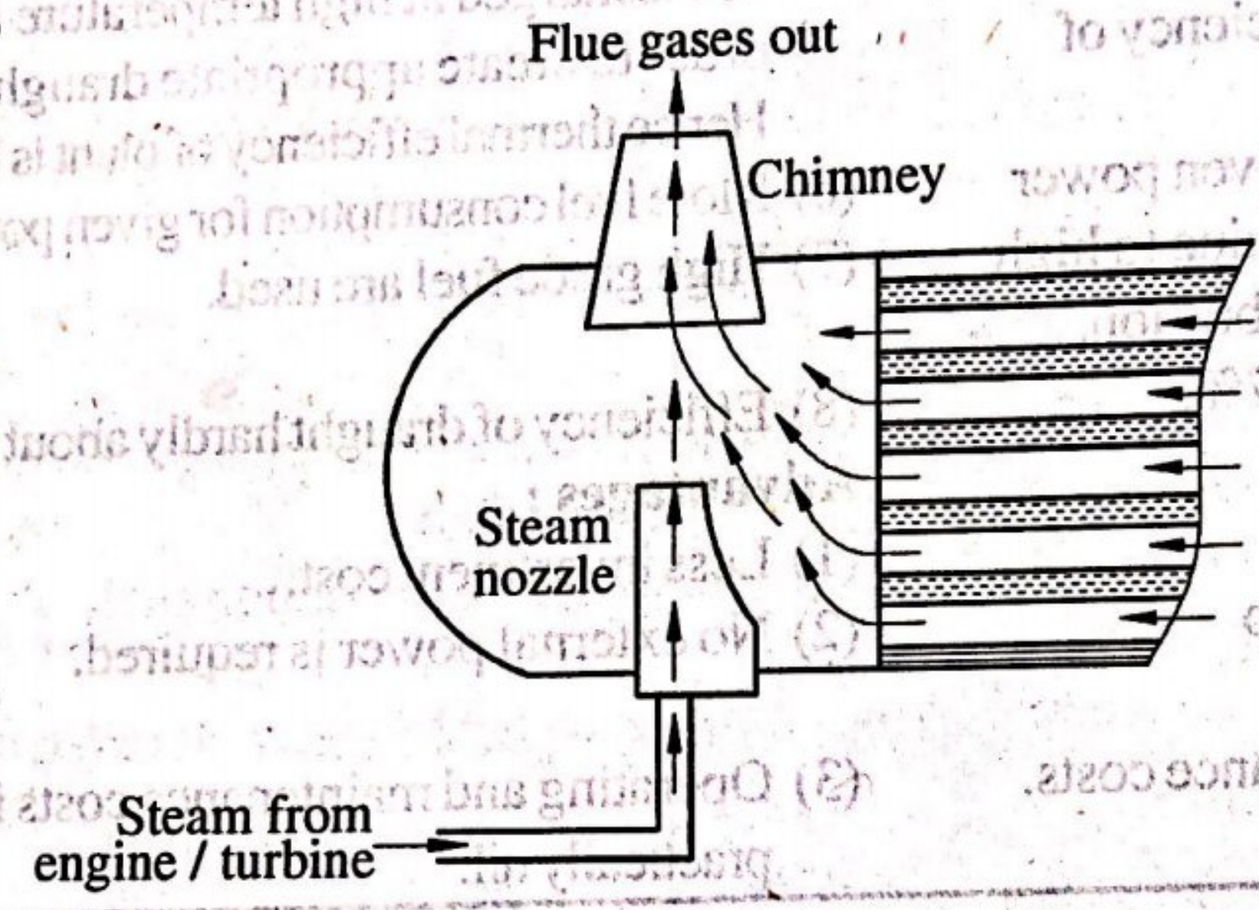


Fig. 4.7 Induced-steam jet draught

In case of induced steam jet draught as shown in Fig. 4.7 the exhaust steam from the engine is forced into smoke box through a nozzle. When the steam flows through the nozzle a partial vacuum is created and the air is induced to the furnace through the flues and grate.

Advantages of steam jet draught :

(1) It is very simple and requires very little attention.
(2) It is very economical with low grade fuel.
(3) Space required is less.

Disadvantages :

(1) It can not be started until steam pressure is available.
(2) In case of forced steam jet draught, steam passing into the furnace, hence it carry away heat with it.
(3) The draught produced is very low.

Exercises

4.1 What is draught ? What are the functions of draught system ?
4.2 Define boiler draught. What are the losses in air-gas circuit of boiler ? Explain in details.
4.3 What do you understand by "Boiler draught"? How the draughts are classified ?
4.4 State the classification of draught.
4.5 State the advantages of natural draught. Explain with sketch the induced steam jet

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draught.

4.6 Discuss the merits and demerits of forced draught over natural draught.

4.7 Prove with usual notations that the draught produced in mm of water head by a chimney is given by

$$h_w = 353H\left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right)\right].$$

4.8 Prove that discharge through chimney will be maximum for the given height of the chimney when $\frac{T_g}{T_a} = 2\left(\frac{m_a+1}{m_a}\right)$.

4.9 What are the limitations of chimney draught ? State the pressure losses which are generally taken into account in designing the draught system.

4.10 Why the height of the chimney and gas exit temperature are important in chimney design ?

4.11 Classify different types of fans. Explain why are centrifugal backward curved blading normally used for forced draft fan.

4.12 Explain what you understand by balanced draught.

4.13 What is the function of chimney in natural draught and artificial draught system ?

4.14 What is the location of forced draught & induced draught fan in boiler draught system ?

4.15 Give advantages of artificial draught over natural draught.

4.16 Discuss the merits and demerits of forced draught over natural draught.

4.17 Derive the expression for fan power and compare power required for F.D. fan and I.D. fan if the air supplied is same both cases.

Unsolved Numerical Problems

4.1 Draught produced by chimney is 15 mm of water column. The height and diameter of chimney are 32 m and 1.6 m respectively. The temperature of flue gases and surrounding air are 310°C and 30°C respectively. Neglect losses and calculate (i) the mass of air induced in boiler per kg of fuel burnt and (ii) mass of flue gases flowing through the chimney in kg/sec. **[Ans :** (i) 6.67 kg/kg of fuel (ii) 28.78 kg/s]

4.2 Calculate the height of chimney to produce a static draught of 20 mm of water column if mean flue gases temperature is 300°C and surrounding air temperature is 27°C. The gas constant for air 287 J/kg K and for gases 255 J/kg K. The atmospheric pressure is 1 bar. **[Ans :** 41.93 m]

4.3 A forced draught fan delivers 1200 m^3 of air per minute through the outlet of 2 m^2 area

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and maintains a static pressure of 110 mm of water. The temperature of air is 21°C. Find the power required to drive the F.D. fan if efficiency of the fan is 75%. Assume that the mass of air at NTP is 1.293 kg per m^3. **[Ans :** 30.376 kW]

4.4 The total draught losses in the boiler passages is 30 mm of water. The velocity of flue gases leaving the chimney is 10 m/s. The flue gases mean temperature and surrounding air temperature are 200°C and 20°C respectively. The mass of air required is 20 kg/ kg of fuel burnt. Calculate power required to drive I.D. pan. **[Ans :** 8.93 kW]

4.5 Compare fan power requirement for ID and FD for following condition :

Temperature of flue gases leaving boiler = 180°C (for both cases)

Temperature of surrounding air = 18°C

Specific heat of flue gases = 1.2 kJ/kg K

Mass of air supplied = 20 kg/kg of fuel burnt

Also compare the quantity of heat is carried away by the flue gases per kg of fuel burnt when the natural draught is used to create same draught, the temperature of flue gases leaving chimney is 300°C and air induced 25 kg/kg of fuel burnt.

[Ans : P_{ID} / P_{FD} = 1.55, Heat carry by flue gases for natural draught = 3460 kJ/kg, Heat carry by flue gases for artificial draught (ID or FD) = 3888 kJ/kg]

Objective Type Questions

1. Boiler draught is defined as
   (a) small pressure difference which causes the flow of steam in the condenser.
   (b) small pressure difference which causes the flow of gases and air inside the boiler.
   (c) the device which removes ash from furnace.
   (d) the device which seperate ash from flue gases.
2. Natural draught produced by given height of chimney depends on
   (a) temperature of flue gases in the chimney
   (b) temperature of boiler surrounding air.
   (c) mass of air required for combustion per unit mass of fuel.
   (d) all of these above
3. The condition of maximum discharge through chimney is given by

   (a) $\frac{T_g}{T_a} = 2\left(\frac{m_a + 1}{m_a}\right)$ (b) $\frac{T_g}{T_a} = 2\left(\frac{m_a}{m_a + 1}\right)$ (c) $\frac{T_g}{T_a} = \left(\frac{m_a}{m_a + 1}\right)$ (d) $\frac{T_g}{T_a} = \left(\frac{m_{a+1}}{m_a}\right)$

4. When height of chimney is equal to the height of a column of hot gases producing draught,

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the draught created by chimney is
(a) minimum (b) maximum (c) zero (d) none of these

5. Draught produced by the chimney alone is
(a) forced draught (b) induced draught
(c) balanced draught (d) none of these

6. Stack refer to a chimney which is made of
(a) rainforced concrete structure (b) metal
(c) masonory (d) none of these

7. The amount of draught created by stack for same height and same flue gases temperature compared to draught produced by brick chimney is
(a) same (b) less (c) more (d) none of these

8. Mechanical draught is created by
(a) fan (b) steam jet (c) stack (d) all of these

9. The draught created by fan which is located at base of chimney is known as
(a) induced draught (b) forced draught
(c) balanced draught (d) steam jet draught

10. In forced draught, the pressure in the furnace is
(a) atmospheric (b) above atmospheric
(c) below atmospheric (d) none of these

11. In balanced draught, the pressure in the furnace is
(a) atmospheric (b) above atmospheric
(c) below atmospheric (d) none of these

12. Power required to drive the induced draught fan as compared to forced draught fan for same draught produced is
(a) equal (b) less (c) more (d) none of these

13. In case of locomotive boilers the draught is produced by
(a) chimney (b) fan (c) steam jet (d) none of these

14. For forced draught fan following type of centrifugal fan generally used due to high tangential speed.
(a) Forward curved blades (b) Backward curved blades
(c) Straight radial blades (d) none of these

Answers

1. (b) **2.** (d) **3.** (a) **4.** (b) **5.** (b) **6.** (b) **7.** (a) **8.** (a) **9.** (a) **10.**(b)
11.(a) **12.** (c) **13.** (c) **14.** (b)

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GTU Study Materials ☑

Chapter 5
Steam Nozzles

5.1 *Introduction*
5.2 *Types of nozzles*
5.3 *Velocity of steam flowing through a nozzle*
5.4 *Friction in a nozzle or Nozzle efficiency*
5.5 *Mass flow rate of steam through the nozzle*
5.6 *Condition for maximum discharge through a Nozzle or Critical pressure ratio or choked flow*
5.7 *Physical significance of critical pressure ratio*
5.8 *General relationship between area, velocity and pressure*
5.9 *Supersaturated flow in nozzles*
5.10 *Effect of variation of back pressure*

5.1 Introduction

The function of a steam nozzle is *to convert the heat energy and pressure energy of steam into kinetic energy*. Its main use is to produce a high velocity jet of steam which is used to drive a steam turbine. This is achieved by allowing the steam to flow (expand) from a region of high pressure at the inlet to a region of low pressure at the outlet. As the steam expands through the nozzles, its velocity and specific volume both increase. Its dryness fraction will also vary due to condensation of steam. Since the weight of steam per second flowing across any section of the nozzle is constant, the cross-section of the nozzle will vary in accordance with velocity, specific volume and the dryness fraction of steam.

A nozzle thus is a device which serves two purposes, namely :

(1) To convert pressure and heat energy into kinetic energy.

(2) To direct the fluid at a specific angle.

5.2 Types of nozzles [Nov. '14]

The nozzles are of three types depending upon its shape.

(1) Convergent nozzle :

When the cross-sectional area of a nozzle decreases continuously from inlet to exit, it is called a convergent nozzle.

In a convergent nozzle, velocity is always less than the sonic or acoustic velocity and hence this nozzle is sometimes called ***subsonic nozzle***. At the smallest section (throat) the velocity becomes sonic velocity. This type of the nozzle is shown in Fig. 5.1(a).

(2) Divergent nozzle :

When the cross-sectional area of a nozzle increases continuously from inlet to exit, it is called a divergent nozzle.

If the velocity of fluid is supersonic i.e. the velocity is more than the sonic velocity and

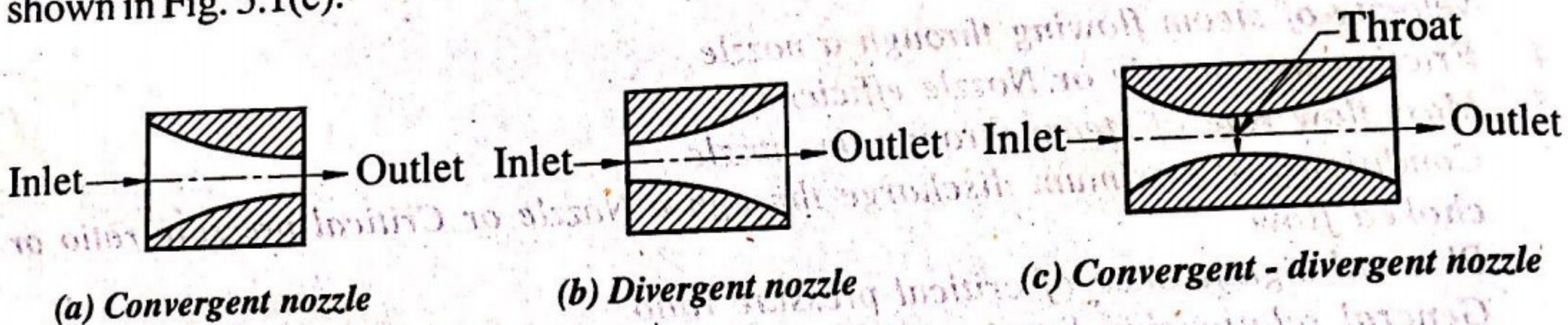
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if we require the velocity to be increased further, then shape of the nozzle is diverging. This shape of the nozzle is shown in Fig. 5.1(b).

(3) Convergent-divergent nozzle :

When the cross-sectional area of a nozzle first decreases from its inlet to throat, and then increases from its throat to exit, it is called a convergent-divergent nozzle as shown in Fig. 5.1(c).



(a) Convergent nozzle *(b) Divergent nozzle* *(c) Convergent - divergent nozzle*

Fig. 5.1 Types of nozzle

In practice, pressure of steam at inlet to the converging portion is very high and the velocity is negligible. Thus, the velocity increases in the converging portion so that the velocity becomes sonic at the throat and further expansion of steam takes place in the diverging portion so that the velocity becomes supersonic in the diverging portion.

5.3 Velocity of steam flowing through a nozzle

Consider the steady flow of unit mass of steam through a nozzle. Let us consider two sections, 1 refers for inlet and 2 refers for exit.

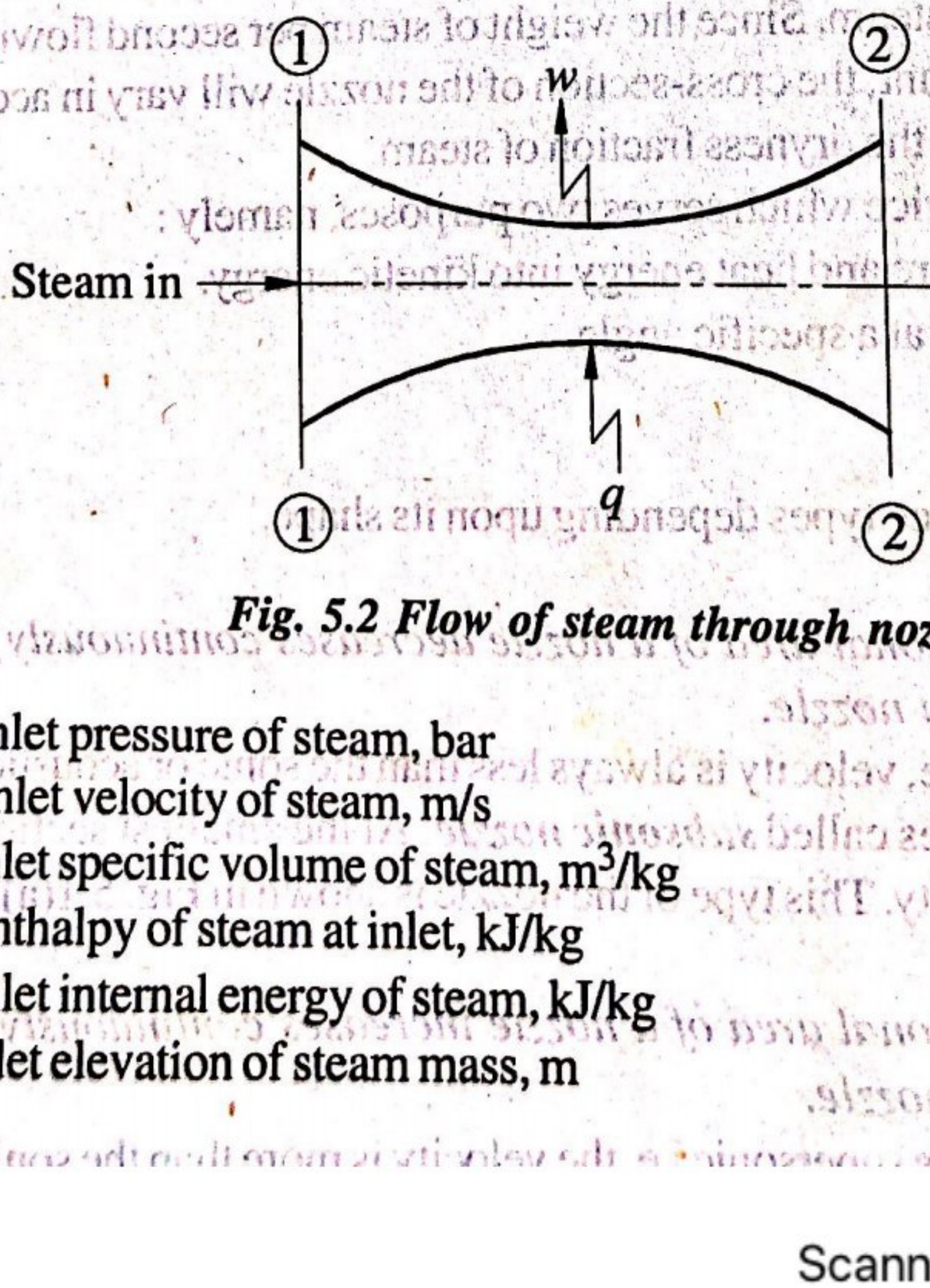


Fig. 5.2 Flow of steam through nozzle

Let

p_1 = inlet pressure of steam, bar
C_1 = inlet velocity of steam, m/s
v_1 = inlet specific volume of steam, m^3/kg
h_1 = enthalpy of steam at inlet, kJ/kg
u_1 = inlet internal energy of steam, kJ/kg
z_1 = inlet elevation of steam mass, m

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w = mechanical work done by nozzle (if any), kJ/kg

q = heat supplied to the nozzle (if any), kJ/kg

p_2, C_2, v_2, h_2, u_2, and z_2 are the corresponding to outlet condition.

Applying the principle of conservation of energy, we can write :

Energy entering into the nozzle = Energy leaving the nozzle

$$p_1v_1 + \frac{C_1^2}{2} + u_1 + gz_1 + q = p_2v_2 + \frac{C_2^2}{2} + gz_2 + w \quad ...(5.1)$$

We know that enthalpy,

$$h_1 = p_1v_1 + u_1, \text{ and } h_2 = p_2v_2 + u_2$$

Now the equation (5.1) becomes

$$h_1 + \frac{C_1^2}{2} + gz_1 + q = h_2 + \frac{C_2^2}{2} + gz_2 + w$$

In nozzle, the change in potential energy is negligible i.e. $z_1 = z_2$

$$\therefore h_1 + \frac{C_1^2}{2} + q = h_2 + \frac{C_2^2}{2} + w$$

Heat is not supplied to the nozzle, i.e. $q = 0$ and there is no mechanical work done by nozzle i.e. $w = 0$

$$\therefore h_1 + \frac{C_1^2}{2} = h_2 + \frac{C_2^2}{2}$$

$$\therefore \frac{C_2^2}{2} = (h_1 - h_2) + \frac{C_1^2}{2}$$

$$\therefore C_2 = \sqrt{2(h_1 - h_2) + C_1^2} \quad ...(5.2)$$

From this equation, we can say that *the gain in kinetic energy between any two section of a nozzle is equal to the loss of enthalpy.*

If inlet velocity of steam to the nozzle is negligible $(C_1 = 0)$, then

$$C_2 = \sqrt{2 \times 10^3 (h_1 - h_2)} \text{ m/s}$$

$$\therefore C_2 = 44.72\sqrt{h_1 - h_2} \text{ m/s} \quad ...(5.3)$$

In actual practice, there is always a certain amount of friction present between the steam and nozzle surfaces. This reduces the heat drop by 10 to 15 percent and thus the exit velocity of steam is also reduced correspondingly. If the flow is with friction, then with friction

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condition value of $(h_1 - h_2)$ will be taken. **[May '14]**

5.4 Friction in a nozzle or Nozzle efficiency

When the steam flows through a nozzle the final velocity of steam for a given pressure drop is reduced due to the following reasons :

(1) The friction between the nozzle surface and steam,

(2) The internal friction of steam itself; and

(3) The shock losses.

The effect of friction on steam flow through a nozzle is represented on Mollier diagram as shown in Fig. 5.3. The diagram can be completed as discussed below :

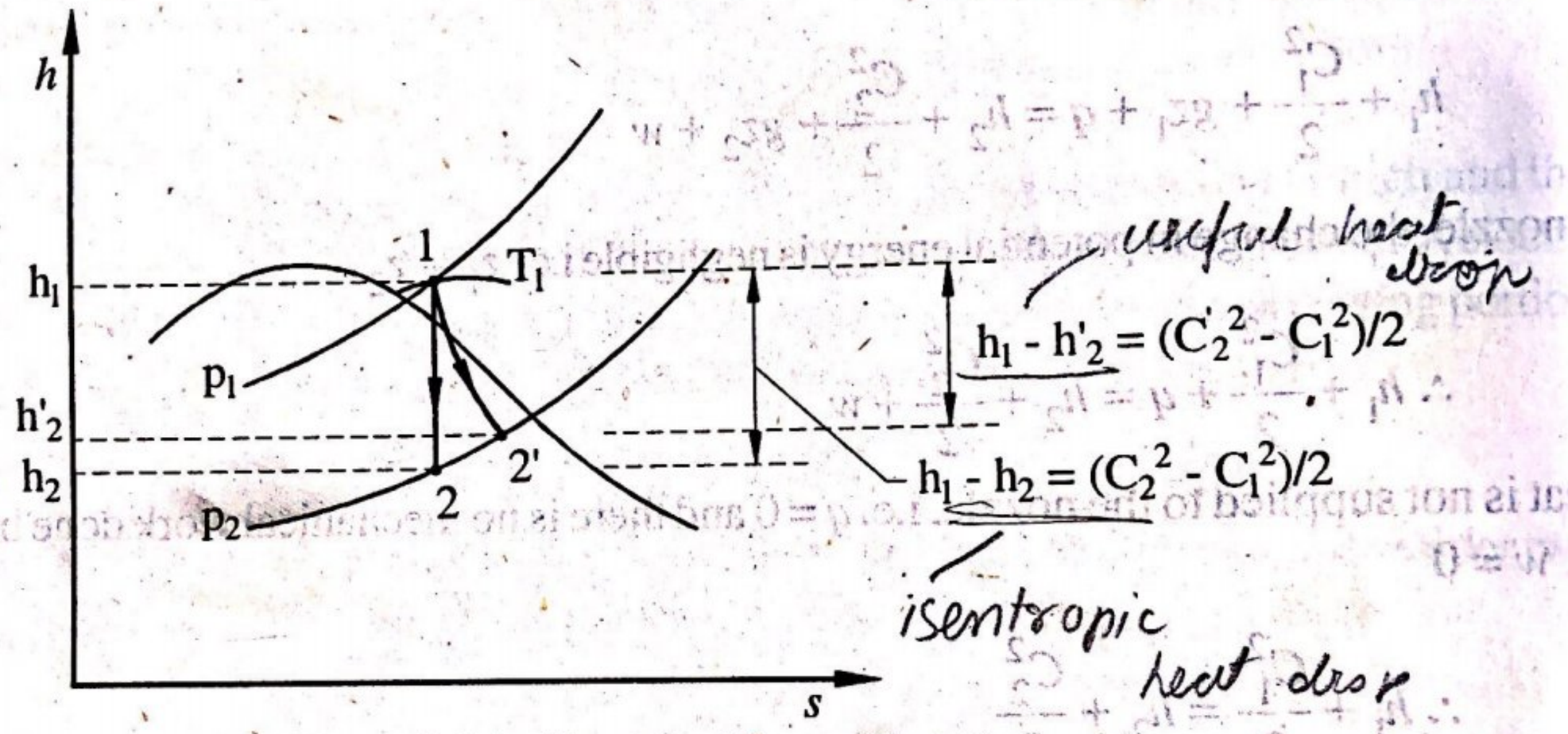


Fig. 5.3 h-s diagram for nozzle efficiency

(1) First of all, locate the point-1 for the initial conditions of the steam. It is a point, where the superheated steam temperature line meets the initial pressure (p_1) line.

(2) Now draw a vertical line through 1 to meet the final pressure p_2 line. This is done as the flow through the nozzle is isentropic, which is expressed by a vertical line 1-2. The heat drop $(h_1 - h_2)$ is known as ***isentropic heat drop***.

(3) Due to friction in the nozzle the actual heat drop in the steam will be less than $(h_1 - h_2)$. Let this heat drop is $(h_1 - h_2')$ instead of $(h_1 - h_2)$.

(4) As the expansion of steam ends at the pressure p_2, therefore final condition of steam is obtained by drawing a horizontal line through h_2' to meet the final pressure p_2 line at $2'$.

(5) Now the actual expansion of steam in the nozzle is expressed by the curve $1 - 2'$ instead of 1–2. The actual heat drop $(h_1 - h_2')$ is known as ***useful heat drop***.

Now nozzle efficiency is defined as *the ratio of useful heat drop to the isentropic heat drop*. Mathematically,

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$$\eta_n = \frac{\text{Useful heat drop}}{\text{Isentropic heat drop}} = \frac{h_1 - h_2'}{h_1 - h_2} = \frac{C_2'^2 - C_1^2}{C_2^2 - C_1^2} \quad ...(5.4)$$

If C_1 is small compared with C_2 then the nozzle efficiency is given by

$$\eta_n = \frac{C_2'^2}{C_2^2} = K_n^2 \quad ...(5.5)$$

$\because C_1 < C_2$

where K_n = Velocity coefficient for the nozzle = C_2' / C_2.

We can say from Fig. 5.3 that the dryness fraction of steam at $2'$ is greater than that at 2. It is thus obvious that the effect of friction is to increase the dryness fraction of steam. This is due to the fact that the energy lost in friction is transferred into heat which tends to dry or superheat the steam.

5.5 Mass flow rate of steam through the nozzle [May '15]

Consider the isentropic flow of steam through a nozzle.

Let p_1 = initial pressure of steam, N/m^2

v_1 = specific volume of steam at pressure p_1, m^3/kg

C_1 = initial steam velocity, m/s

A_2 = Cross sectional area of nozzle at outlet, m^2, where the pressure is p_2.

p_2, v_2 and C_2 are the corresponding to outlet condition.

Fig. 5.4 shows p-v diagram for isentropic expansion of steam through nozzle.

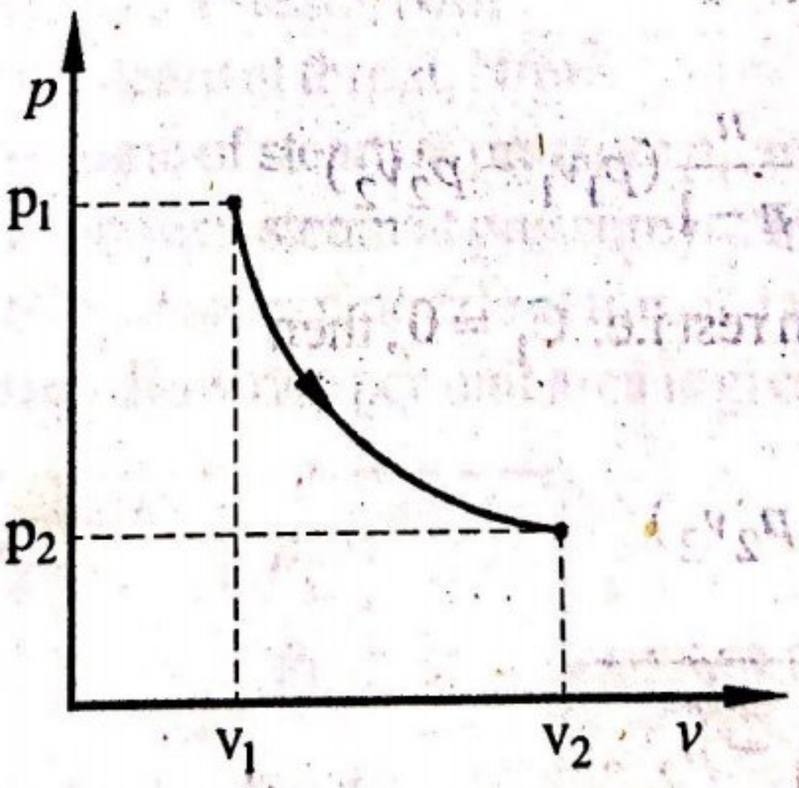


Fig. 5.4 p-v diagram for isentropic expansion

The isentropic flow of steam through the nozzle may approximately be represented by equation.

$$pv^n = \text{constant} = c$$

where, $n = 1.135$ for dry and saturated steam, and

$= 1.3$ for superheated steam.

For wet steam, the value of n can be calculated by Zenner's equation, $n = 1.035 +$

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0.1x, where x is the initial dryness fraction of steam.

These values of n are approximate only, actually the value of n varies during the expansion.

$$v = \left(\frac{c}{p}\right)^{1/n}$$

$$\therefore\ c^{1/n} = vp^{1/n}, \qquad \therefore\ v = c^{1/n} \cdot p^{-1/n}$$

The gain in kinetic energy of fluid flowing through the nozzle is given by

Gain in kinetic energy = Adiabatic heat drop

= work done during cycle

$$\frac{C_2^2 - C_1^2}{2} = \int_{p_2}^{p_1} v\,dp$$

$$\int_{p_2}^{p_1} v\,dp = \int_{p_2}^{p_1} c^{1/n} p^{-1/n}\,dp = c^{1/n} \int_{p_2}^{p_1} p^{-1/n}\,dp$$

$$= \frac{n}{n-1} c^{1/n} (p_1^{(n-1)/n} - p_2^{(n-1)/n})$$

$$= \frac{n}{n-1}(p_1v_1 - p_2v_2) \quad (\because c^{1/n} = v_1 p_1^{1/n} = v_2 p_2^{1/n})$$

$$\therefore \frac{C_2^2 - C_1^2}{2} = \frac{n}{n-1}(p_1v_1 - p_2v_2) \qquad \ldots(5.6)$$

If expansion of steam is from rest i.e. $C_1 = 0$, then

$$\frac{C_2^2}{2} = \frac{n}{n-1}(p_1v_1 - p_2v_2)$$

$$\therefore \boxed{C_2 = \sqrt{\frac{2n}{n-1}(p_1v_1 - p_2v_2)}}$$

But $p_1v_1^n = p_2v_2^n \qquad \therefore v_2 = v_1\left(\frac{p_2}{p_1}\right)^{-1/n}$

$$\therefore \boxed{C_2 = \sqrt{\frac{2n}{n-1} p_1v_1\left(1 - \left(\frac{p_2}{p_1}\right)^{(n-1)/n}\right)}} \qquad \ldots(5.7)$$

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From the continuity equation the mass flow of steam in *kg. per second* through the cross-sectional area A_2 and at pressure p_2 is given by

$$m = \frac{A_2 C_2}{v_2} \qquad ...(5.8)$$

Substituting the value of C_2 and v_2 into the equation (5.8), we get

$$m = \frac{A_2}{v_1 (p_2 / p_1)^{-1/n}} \sqrt{\frac{2n}{n-1} p_1 v_1 \left(1 - \left(\frac{p_2}{p_1}\right)^{(n-1)/n}\right)}$$

$$\therefore m = A_2 \sqrt{\frac{2n}{n-1} \frac{p_1}{v_1} \left[\left(\frac{p_2}{p_1}\right)^{2/n} - \left(\frac{p_2}{p_1}\right)^{(n+1)/n}\right]} \qquad ...(5.9)$$

This is a very important relation which gives mass flow rate in kg/s.

5.6 Condition for maximum discharge through a Nozzle or Critical pressure ratio or choked flow. [May '14, May '13, May '12]

*The pressure at which the area is a minimum and the discharge per unit area a maximum is termed as the **critical pressure**.*

A nozzle is, normally, designed for maximum discharge by designing a certain throat pressure (critical pressure) which produces this condition.

Let, p_1 = initial pressure of steam, N/m^2

p_2 = pressure of steam at throat, N/m^2

v_1 = specific volume of steam at pressure p_1, m^3/kg

v_2 = specific volume of steam at pressure p_2, m^3/kg.

A_2 = cross sectional area of nozzle at throat, m^2

From the equation (5.9), the mass flow rate per unit area is given by

$$\frac{m}{A_2} = \sqrt{\frac{2n}{n-1} \frac{p_1}{v_1} \left[\left(\frac{p_2}{p_1}\right)^{2/n} - \left(\frac{p_2}{p_1}\right)^{(n+1)/n}\right]} \qquad ...(5.10)$$

In the above equation, p_1, v_1, and n are all constants, hence the mass flow rate per unit area is a function of the ratio of p_2 to p_1. There will be only one value of ratio p_2/p_1 which will produce maximum discharge for the nozzle. Therefore, only that portion of the equation which contains p_2/p_1 is differentiated and equated to zero for maximum discharge.

$$\frac{d}{d\left(\frac{p_2}{p_1}\right)} \left[\left(\frac{p_2}{p_1}\right)^{2/n} - \left(\frac{p_2}{p_1}\right)^{(n+1)/n}\right] = 0$$

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Let $\frac{p_2}{p_1} = r$

$$\therefore \frac{d}{dr}[r^{2/n} - r^{(n+1)/n}] = 0$$

$$\therefore \frac{2}{n} r^{(2/n)-1} - \frac{n+1}{n} r^{\frac{n+1}{n}-1} = 0$$

Dividing eq^n by $(2/n)$

$$\therefore r^{\frac{2-n}{n}} - \frac{n+1}{2} r^{\frac{1}{n}} = 0$$

$$\therefore r^{\frac{2-n}{n}} \times r^{-\frac{1}{n}} = \frac{n+1}{2}$$

$$\therefore r^{\frac{2-n}{n}-\frac{1}{n}} = \frac{n+1}{2}; \quad \therefore r^{\frac{1-n}{n}} = \frac{n+1}{2}$$

$$\therefore r = \left(\frac{n+1}{2}\right)^{\frac{n}{1-n}} = \left(\frac{n+1}{2}\right)^{\frac{-n}{-(1-n)}}$$

$$\therefore r = \frac{p_2}{p_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \quad \ldots(5.11)$$

The ratio p_2/p_1 is known as critical pressure ratio and depends on the value of n.

The pressure p_2 at the throat is known as critical pressure.

We know that for dry saturated steam, $n = 1.135$

$$\therefore \frac{p_2}{p_1} = \left(\frac{2}{1.135+1}\right)^{\frac{1.135}{1.135-1}} = 0.5774 \cong 0.58, \quad \therefore p_2 = 0.58 p_1$$

For superheated steam, $n = 1.3$

$$\therefore \frac{p_2}{p_1} = \left(\frac{2}{1.3+1}\right)^{\frac{1.3}{1.3-1}} = 0.5457 \cong 0.545, \quad \therefore p_2 = 0.545 p_1$$

The maximum value of the mass flow rate per unit area per second is obtained by substituting the value of p_2/p_1 in equation (5.10).

∴ Maximum discharge,

$$\left(\frac{m}{A_2}\right)_{max} = \sqrt{\frac{2n}{n-1}\frac{p_1}{v_1}\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$$

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$$= \sqrt{\frac{2n}{n-1}\frac{p_1}{v_1}\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}}\left[1-\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}-\frac{2}{n-1}}\right]}$$



$$= \sqrt{\frac{2n}{n-1}\frac{p_1}{v_1}\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}}\left[1-\left(\frac{2}{n+1}\right)\right]}$$

$$= \sqrt{\frac{2n}{n-1}\frac{p_1}{v_1}\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}}\left[\frac{n-1}{n+1}\right]}$$

$$\left(\frac{m}{A_2}\right)_{max} = \sqrt{\frac{2n}{n+1}\frac{p_1}{v_1}\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}}}$$

$$\therefore m_{max} = A_2\sqrt{\frac{2n}{n+1}\frac{p_1}{v_1}\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}}} \quad ...(5.12)$$

From the above equation, it is clear that *in a convergent-divergent nozzle, the discharge depends upon the area of nozzle at throat and the initial conditions of the steam* (i.e. pressure p_1 and volume v_1). It is independent of the exit conditions of the steam.

(1) When the steam is initially dry saturated :

We know that for dry saturated steam, $n = 1.135$. Therefore substituting the value of n in the relation for maximum discharge, we have

$$m_{max} = 0.637 A_2\sqrt{\frac{p_1}{v_1}}$$

(2) When the steam is initially superheated :

We know that for superheated steam, $n = 1.3$. Therefore, substituting the value of n in the relation for maximum discharge. We have

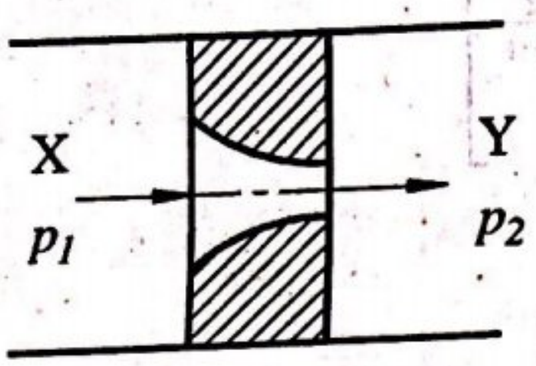
$$m_{max} = 0.666 A_2\sqrt{\frac{p_1}{v_1}}$$

5.7 Physical significance of critical pressure ratio

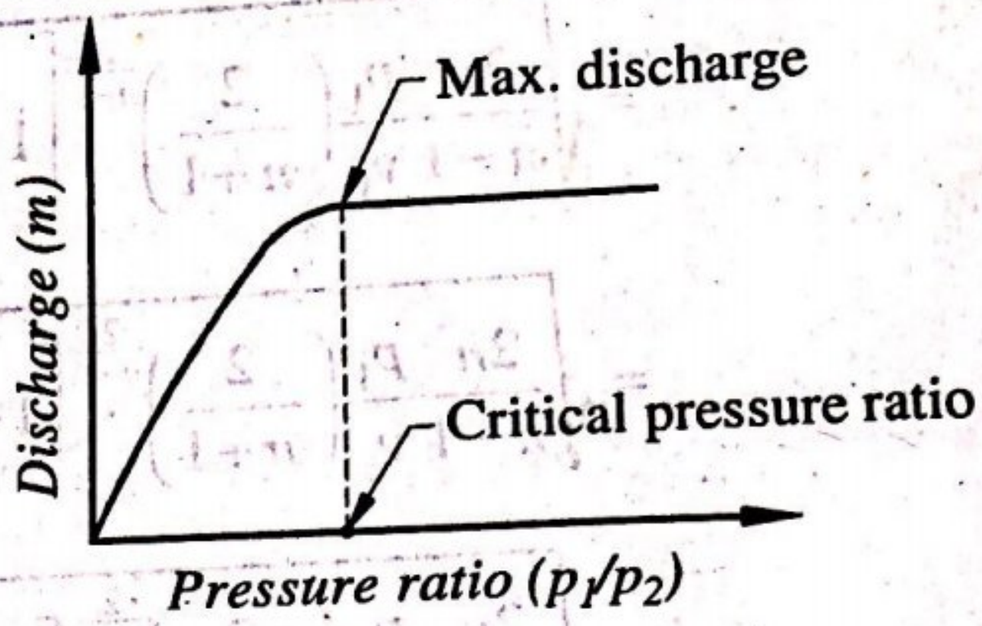
Now consider two vessels X and Y connected by a convergent nozzle as shown in Fig. 5.5(a). Let the vessel X contains steam at a high and steady pressure (p_1), and the vessel Y contains steam at another pressure (p_2) which may be varied at will.

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(a) Nozzle section



(b) Discharge - pressure ratio curve

Fig. 5.5

Let the pressure p_2 in vessel Y be initially equal to p_1, then there is no flow of steam through the nozzle. When pressure p_2 is reduced gradually, the discharge (m) through the nozzle will increase as shown by the curve in Fig. 5.5(b). As the pressure p_2 approaches the critical pressure, the discharge rate gradually approaches its maximum value and when p_2 is reduced below the critical value, the discharge rate remains constant. Thus we see that the reduction in pressure p_2 below the critical value does not increase the rate of discharge.

We know that the velocity of steam at any section in the nozzle (equation (5.7))

$$C_2 = \sqrt{\frac{2n}{n-1} p_1 v_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}$$

and the critical pressure ratio for maximum discharge,

$$\frac{p_2}{p_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \quad \text{or} \quad \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = \frac{2}{n+1}$$

$$\therefore C_2 = \sqrt{\frac{2n}{n-1} p_1 v_1 \left[1 - \frac{2}{n+1}\right]} = \sqrt{\frac{2n}{n-1} p_1 v_1 \left[\frac{n+1-2}{n+1}\right]}$$

$$\therefore C_2 = \sqrt{\frac{2n}{n+1} p_1 v_1} \qquad ...(5.13)$$

The velocity at the throat with isentropic expansion from rest may also be expressed in terms of the final pressure p_2 and specific volume v_2. As we know the velocity of steam at throat is

$$C_2 = \sqrt{\frac{2n}{n-1}(p_1 v_1 - p_2 v_2)}$$

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$$= \sqrt{\frac{2n}{n-1} p_2 v_2 \left[\left(\frac{p_2}{p_1} \right)^{\frac{1-n}{n}} - 1 \right]}$$

Substituting the value of $\frac{p_2}{p_1} = \left(\frac{2}{n+1} \right)^{\frac{n}{n-1}}$, we get

$$C_2 = \sqrt{\frac{2n}{n-1} p_2 v_2 \left[\frac{n+1}{2} - 1 \right]} = \sqrt{\frac{2n}{n-1} p_2 v_2 \frac{n-1}{2}}$$

$$\therefore C_2 = \sqrt{n p_2 v_2} \quad \text{...(5.14)}$$

This equation is a very important deduction. This is the value of velocity of sound in the medium (steam) at pressure p_2 and is known as ***sonic velocity***.

From the above discussion it follows that the maximum mass flow through a convergent nozzle is obtained when the pressure ratio across the nozzle is the critical pressure ratio. The flow in the convergent portion of the nozzle is sub-sonic and in the divergent portion it is supersonic. *When a nozzle operates with the maximum mass flow rate, it is said to be* ***choked***. A correctly designed convergent-divergent nozzle is always chocked.

To increase the velocity of steam above sonic velocity (supersonic) by expanding steam below the critical pressure, the divergent portion for the nozzle is necessary.

Problem 5.1 *Steam is expanded in a nozzle from 15 bar and 350°C to 5 bar. Find the throat and exit area if the flow rate is 1 kg/s. What should be the coefficient of velocity if the exit velocity is 650 m/s ?*

Solution : Given data

$p_1 = 15$ bar, $T_1 = 350°C$, $p_3 = 5$ bar, $m = 1$ kg/s, $(C_3)_{actual} = 650$ m/s, $A_2 = ?$, $A_3 = ?$, $K_n = ?$

Refer Fig. 5.6. Let suffix 1, 2 and 3 represent the inlet, throat, and exit conditions of nozzle.

From Mollier's diagram

At 15 bar and 350°C, $h_1 = 3150$ kJ/kg

The throat pressure (critical pressure) for superheated steam is calculated by

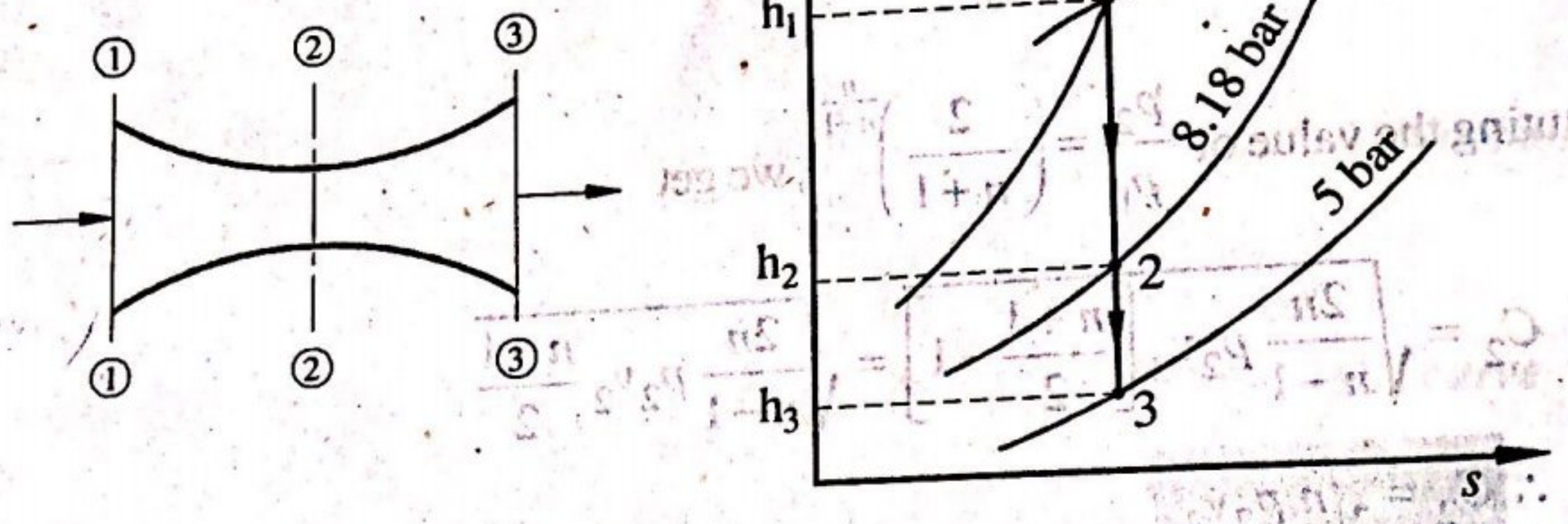
$$p_2 = p_1 \left(\frac{2}{n+1} \right)^{\frac{n}{n-1}} = 0.5457\, p_1$$

$$\therefore p_2 = 0.5457 \times 15 = 8.1855 \text{ bar}$$

Since the throat pressure is more than back pressure, hence the nozzle is choking and a convergent-divergent nozzle is required.

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Nozzle *h - s diagram*

Fig. 5.6

From Mollier chart, for isentropic expansion

$T_2 = 275°C, \quad h_2 = 2990$ kJ/kg.

Therefore, velocity at the throat is given by

$$C_2 = 44.72\sqrt{h_1 - h_2} = 44.72\sqrt{3150 - 2990} = 565.67 \text{ m/s}$$

From superheated steam tables at 8.18 bar and 275°C, $v_2 = 0.3086$ m³/kg

Area at the throat is given by

$$A_2 = \frac{mv_2}{C_2} = \frac{1 \times 0.3086}{565.67} = 5.455 \times 10^{-4} m^2 = \mathbf{5.455 \text{ cm}^2} \quad \textit{Ans}$$

For finding out exit area, the velocity and the specific volume of steam at the exit are required.

At 5 bar, $h_3 = 2890$ kJ/kg, $T_3 = 200°C$, $v_3 = 0.4250$ m³/kg

Neglecting inlet velocity of steam, $C_1 = 0$

Exit velocity, $C_3 = 44.72\sqrt{h_1 - h_3}$

$$= 44.72\sqrt{3150 - 2890} = 721.088 \text{ m/s}$$

Exit area is given by

$$A_3 = \frac{mv_3}{C_3} = \frac{1 \times 0.4250}{721.088} = 5.894 \times 10^{-4} m^2 = \mathbf{5.894 \text{ cm}^2} \quad \textit{Ans}$$

Nozzle efficiency, $\eta_n = K_n^2 = \dfrac{(C_3)^2_{actual}}{(C_3)^2_{isen}}$

$$= \left(\frac{650}{721.088}\right)^2$$

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$\therefore K_n = 0.9014$ **Ans**

Problem 5.2 *A nozzle expands steam from 12 bar and 250°C to 6 bar. Is the nozzle convergent or convergent divergent ? Neglecting the initial velocity, find the minimum area of the nozzles to flow 2 kg/s of steam under the given conditions. Assume the expansion of steam is isentropic. Calculate the actual throat area if the coefficient of discharge is 0.98.* **[Same as May '12]**

Solution : Given data (Refer Fig. 5.7)

$p_1 = 12$ bar, $T_1 = 250°C$, $p_3 = 6$ bar, $m = 2$ kg/s, $A_2 = ?$ $(A_2)_{actual} = ?$, $C_d = 0.98$

Let 1, 2 and 3 represent initial, throat and exit conditions respectively.

The throat (critical) pressure for superheated steam is given by

$$p_2 = p_1\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} = 0.5457\,p_1 = 0.5457\times 12 = 6.55\,\text{bar}$$

Since the exit pressure is 6 bar which is less than critical pressure so the nozzle is **convergent-divergent.** ***Ans***

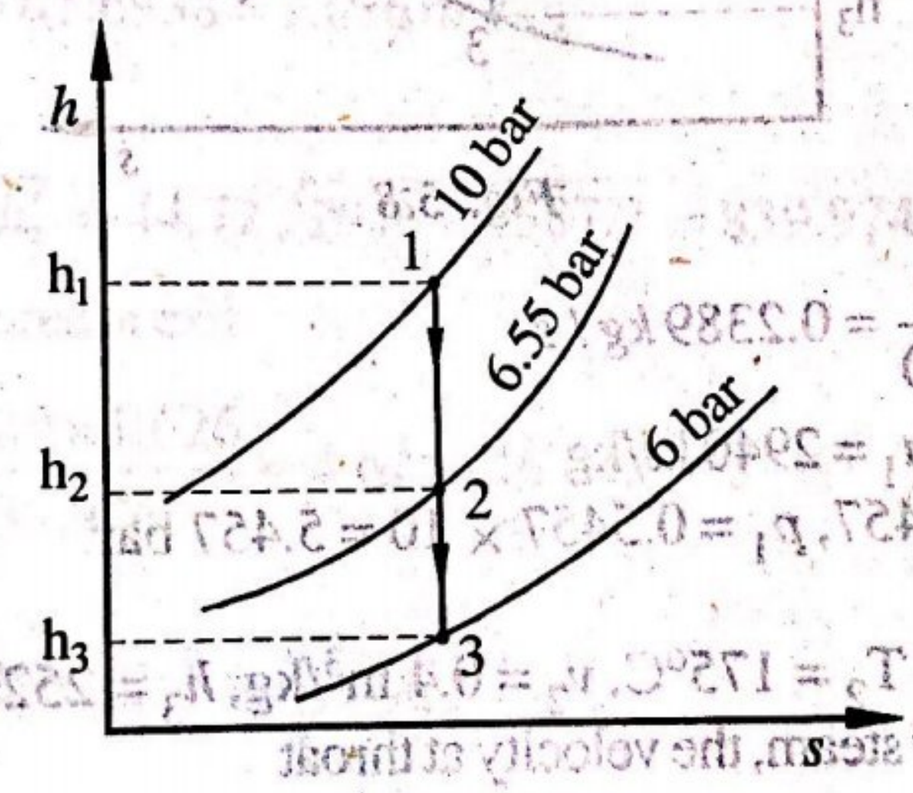


Fig. 5.7

From Mollier's chart, for isentropic expansion

$h_1 = 2930$ kJ/kg, $h_2 = 2810$ kJ/kg, $T_2 = 180°C$, $v_2 = 0.350\ m^3/kg$.

The velocity at the throat is given by

$$C_2 = 44.72\sqrt{h_1 - h_2} = 44.72\sqrt{2930 - 2810} = 489.88\,m/s$$

The minium area is given by

$$A_2 = \frac{mv_2}{C_2} = \frac{2\times 0.350}{489.88} = 1.429\times 10^{-3}\,m^2 = \mathbf{14.29\ cm^2}$$ ***Ans***

The actual throat area is given by

$$(A_2)_{actual} = \frac{A_2}{C_d} = \frac{14.29}{0.98} = \mathbf{14.581\ cm^2}$$ ***Ans***

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***Problem** 5.3 A steam is expanded in convergent-divergent nozzle from a pressure of 10 bar and superheated to 250°C to 1 bar. Friction loss in the diverging part of the nozzle may be taken as 0.16 of the total enthalpy drop. Calculate the throat and exit diameters of the nozzle which will discharge 860 kg of steam per hour.*

Solution : Given data (Refer Fig. 5.8)

$p_1 = 10$ bar, $T_1 = 250°C$, $p_3 = 1$ bar, $D_2 = ?$, $D_3 = ?$

Let 1, 2 and 3 represent initial, throat and final conditions of steam in the nozzle.

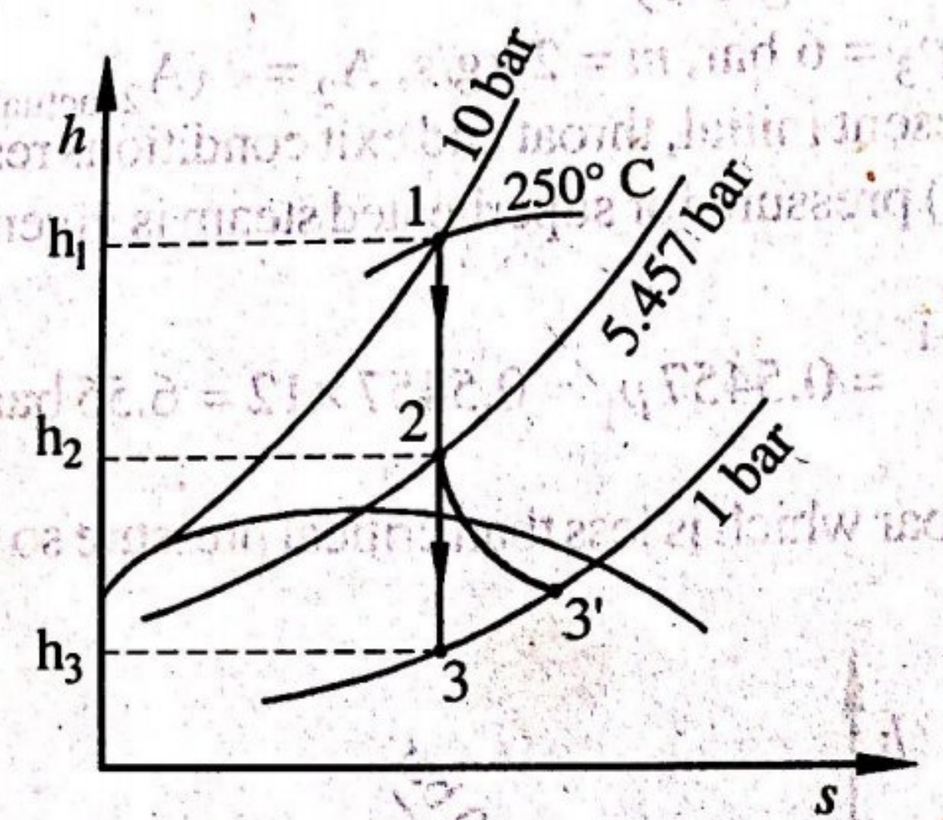


Fig. 5.8

Mass flow rate, $m = \dfrac{860}{3600} = 0.2389\ kg/s$

From Mollier's diagram, $h_1 = 2940$ kJ/kg

Throat pressure, $p_2 = 0.5457$, $p_1 = 0.5457 \times 10 = 5.457$ bar

From Mollier's diagram,

$h_2 = 2830$ kJ/kg, $T_2 = 175°C$, $v_2 = 0.4\ m^3/kg$, $h_3 = 2520$ kJ/kg.

Neglecting inlet velocity of steam, the velocity at throat

$$C_2 = 44.72\sqrt{h_1 - h_2} = 44.72\sqrt{2940 - 2830}$$

$$= 469.027 \text{ m/s}$$

From continuity equation, the throat area

$$A_2 = \frac{mv_2}{C_2} = \frac{0.2389 \times 0.4}{469.027} = 2.037 \times 10^{-4}\ m^2$$

Also $A_2 = \dfrac{\pi}{4} D_2^2$

$$\therefore\ 2.037 \times 10^{-4} = \frac{\pi}{4} D_2^2$$

$$D_2 = 0.0161\ m = 1.61\ cm$$ **Ans**

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Alternatively, throat area may also be found out from the equation for n = 1.3 since throat condition is in superheated zone.

$$m = 0.666 A_2 \sqrt{\frac{p_1}{v_1}}$$

It is assumed that friction loss in the diverging part of the nozzle is taken as 0.16 of the total isentropic enthalpy drop i.e. $\eta_n = 0.84$

$$\therefore h_3' - h_3 = \text{friction loss} = 0.16(h_1 - h_3)$$

$$\therefore h_3' - 2520 = 0.16(2940 - 2520)$$

$$\therefore h_3' = 2587.2\, kJ/kg$$

From chart, $x_3' = 0.96$,

From steam tables at 1 bar, $v_{g3} = 1.6938\ m^3/kg$

$$v_3' = x_3' v_{g3} = 0.96 \times 1.6938 = 1.626\ m^3/kg$$

The actual velocity at exit

$$C_3' = 44.72\sqrt{h_1 - h_3'} = 44.72\sqrt{2940 - 2587.2} = 839.974\ m/s$$

From continuity equation, area at exit

$$A_3' = \frac{m v_3'}{C_3'} = \frac{0.2389 \times 1.626}{839.974} = 4.625 \times 10^{-4}\ m^2$$

$$A_3' = \frac{\pi}{4} D_3^2 = 4.625 \times 10^{-4} = \frac{\pi}{4} \times D_3^2$$

$$\therefore D_3 = 0.02427\ m = \mathbf{2.427\ cm} \quad \textit{Ans}$$

Problem 5.4 *Estimate the mass flow rate of steam in a nozzle with the following data :*
Inlet pressure and temperature = 12 bar and 200°C
Back pressure = 1 bar
Throat diameter = 10 mm ***[G.U. Dec. 06, 6 marks]***

Solution : Given data :

$p_1 = 12$ bar, $T_1 = 200°C$, $p_3 = 1$ bar, $D_2 = 10$ mm $= 0.01$ m, $m = ?$

For initially superheated steam, throat pressure is given by

$$p_2 = 0.5457 \times p_1 = 0.5457 \times 12 = 6.548\ bar$$

Since the throat pressure is more than back pressure, hence the nozzle is chocking

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and a convergent-divergent nozzle is required.

From Mollier chart, for isentropic expansion

$$h_1 = 2820 \text{ kJ/kg},\ h_2 = 2690 \text{ kJ/kg},\ x_2 = 0.97$$

From steam tables, $v_{g_2} = 0.28829$

Velocity at the throat is given by

$$C_2 = 44.72\sqrt{h_1 - h_2} = 44.72\sqrt{2820 - 2690} = 509.89 \text{ m/s}$$

Area at the throat is given by

$$A_2 = \frac{mv_2}{C_2};\ \frac{\pi}{4}D_2^2 = \frac{m \times x_2 v_{g2}}{C_2}$$

$$\therefore \frac{\pi}{4}(0.01)^2 = \frac{m \times 0.97 \times 0.28829}{509.89}$$

$$\therefore\ m = \mathbf{0.1432 \text{ kg/s}}$$ **Ans**

Problem 5.5 *A steam nozzle is to be designed for a mass flow rate of 8 kg/s of steam from 8 bar and 350°C to a back pressure of 1 bar. The nozzle efficiency is 0.93 and the frictional loss is assumed to take place in the diverging portion of the nozzle only. Assume a critical pressure ratio of 0.5457. Determine the throat and exit area.* ***[May '15]***

Solution : Given data : (Refer Fig. 5.9)

$m = 8$ kg/s, $p_1 = 8$ bar, $T_1 = 350°C$, $p_3 = 1$ bar, $\eta_n = 0.93$, $p_2 / p_1 = 0.5457$,

$A_2 = ?,\ A_3 = ?$

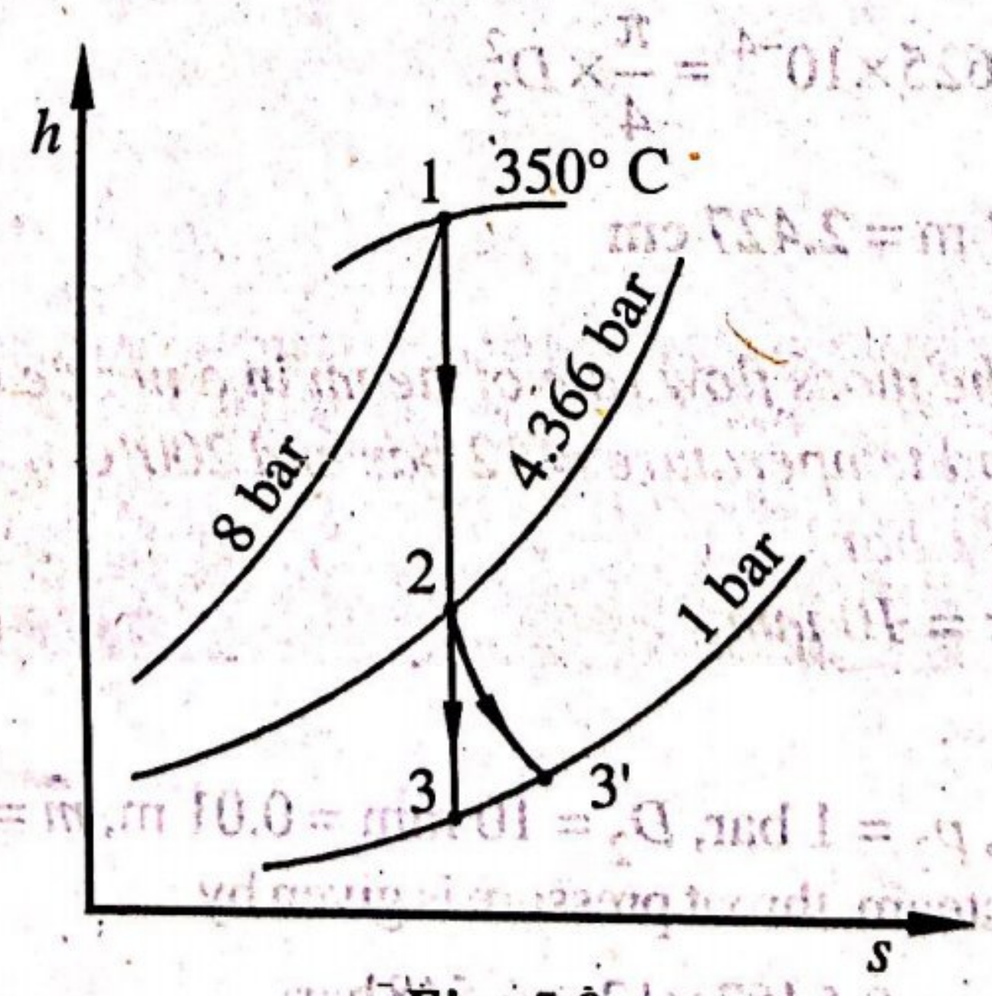


Fig. 5.9

Let 1, 2, and 3 represent inlet, throat and exit condition of the nozzle. From Mollier's diagram.

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$h_1 = 3160$ kJ/kg, $p_2 = 0.5457\, p_1 = 0.5457 \times 8 = 4.366$ bar

The vertical line drawn from point 1 to pressure line p_2 gives the state point 2.

$\therefore h_2 = 2990$ kJ/kg, $T_2 = 265°C$, $v_2 = 0.485$ m³/kg, $h_3 = 2710$ kJ/kg

The velocity of steam at throat is given by

$$C_2 = 44.72\sqrt{h_1 - h_2} = 44.72\sqrt{3160 - 2990} = 583.077 \text{ m/s}$$

The throat area is given by

$$A_2 = \frac{mv_2}{C_2} = \frac{8 \times 0.485}{583.077} = 6.654 \times 10^{-3}\, m^2 = \mathbf{66.54\ cm^2} \qquad \textit{Ans}$$

We have, nozzle efficiency, $\eta_n = \dfrac{h_1 - h_3'}{h_1 - h_3}$

$$0.93 = \frac{3160 - h_3'}{3160 - 2710}$$

$$\therefore h_3' = 2741.5 \text{ kJ/kg}$$

The actual exit velocity is given by

$$C_3' = 44.72\sqrt{h_1 - h_3'} = 44.72\sqrt{3160 - 2741.5} = 914.85 \text{ m/s}$$

From the Mollier's diagram at point 3′, $v_3' = 1.9$ m³/kg

The actual area at the exit is given by

$$A_3' = \frac{mv_3'}{C_3'} = \frac{8 \times 1.9}{914.85} = 0.016615 \text{ m}^2 = \mathbf{166.15\ cm^2} \qquad \textit{Ans}$$

Problem 5.6 *A steam turbine develops 185 kW with a consumption of 16.5 kg/kWh. The pressure and temperature of steam entering the nozzle are 12 bar and 220°C respectively. The steam leaves the nozzle at 1.2 bar. If the diameter of the nozzle at throat is 7 mm, find the minimum number of nozzles required.* **[G.U., Dec. '05, 8 Marks]**

Solution : Given data

$P = 185$ kW, $m_s = 16.5$ kg/kwh, $p_1 = 12$ bar, $T_1 = 220°C$, $p_3 = 1.2$ bar, $D_2 = 7$ mm,

Number of nozzles = ?

Let 1, 2 and 3 represent inlet, throat and exit conditions of the nozzle.

For superheated steam, throat pressure is given by

$$p_2 = 0.5457\, p_1 = 0.5457 \times 12 = 6.548 \text{ bar}$$

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The Mollier's diagram for the expansion of steam through the nozzle is shown in Fig. 5.10.

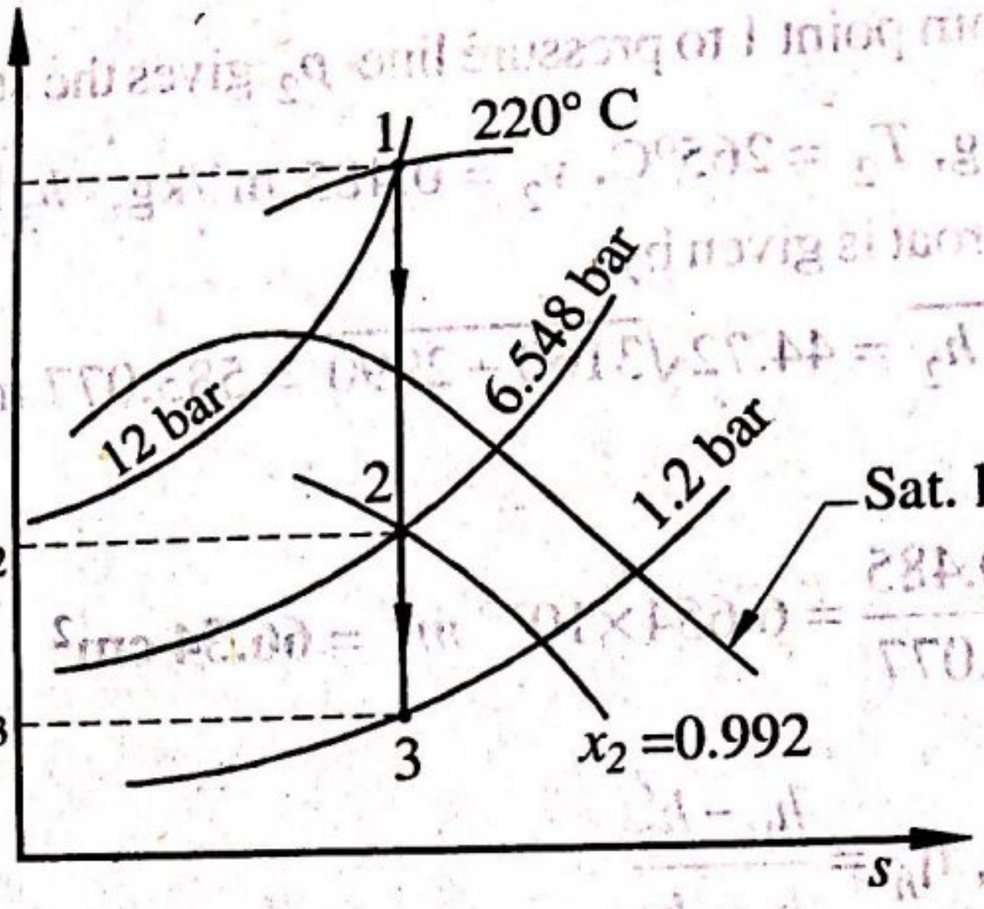


Fig. 5.10

From the Mollier's diagram, we find that enthalpy of steam at entrance,

$$h_1 = 2860 \text{ kJ/kg}$$

Enthalpy of steam at throat, $h_2 = 2750$ kJ/kg and dryness fraction of steam at throat,

$$x_2 = 0.992.$$

From steam tables at $p_2 = 6.548$ bar pressure, $v_{g_2} = 0.29 \text{ m}^3/\text{kg}$

$\therefore$ Specific volume of steam at throat, $v_2 = x_2 v_{g_2} = 0.992 \times 0.29 = 0.2877 \text{ m}^3/\text{kg}$

Velocity of steam at throat,

$$C_2 = 44.72\sqrt{h_1 - h_2} = 44.72\sqrt{2860 - 2750} = 470 \text{ m/s}$$

Area of nozzle at throat,

$$A_2 = \frac{\pi}{4} D_2^2 = \frac{\pi}{4} \times (7 \times 10^{-3})^2 = 38.5 \times 10^{-6} m^2$$

Mass flow rate per nozzle,

$$m = \frac{A_2 C_2}{v_2} = \frac{38.5 \times 10^{-6} \times 470}{0.2877} = 0.063 \, kg/s$$

We know that total mass flow rate

$$= 16.5 \times 185 = 3052.5 \, kg/h = 0.848 \, kg/s$$

$$\therefore \text{ Number of nozzles} = \frac{\text{Total mass flow rate}}{\text{Mass flow rate per nozzle}} = \frac{0.848}{0.063} = 13.46 \text{ say } \mathbf{14} \quad \textit{Ans}$$

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5.8 General relationship between area, velocity and pressure

Let us consider the steady, adiabatic and frictionless flow through nozzle. In Fig. 5.11, we have two transverse plane section a distance dx apart. We assume that (i) the nozzle runs full, and (ii) the velocity is uniform across any section. Then by the continuity equation, we have

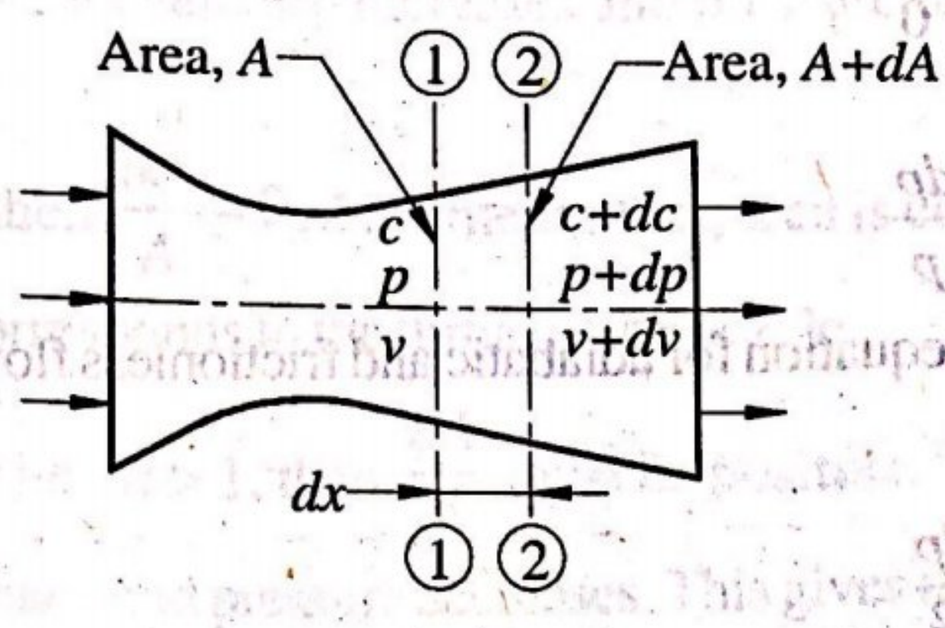


Fig. 5.11

$$m = \frac{AC}{v} = \frac{(A+dA)(C+dC)}{v+dv}$$

$$\therefore (A+dA)(C+dC)v = AC(v+dv)$$

$$\therefore (AC + AdC + CdA + dAdC)v = ACv + ACdv$$

Since the product $dAdC$ is very small, so it may be neglected, then

$$(AC + AdC + CdA)v = ACv + ACdv$$

$$\therefore AvdC + vCdA = ACdv$$

Dividing both sides by ACv, we have

$$\frac{dC}{C} + \frac{dA}{A} - \frac{dv}{v} = 0 \quad \text{....(5.15)}$$

This equation may also be derived by following method $\frac{AC}{v} = m =$ constant.

Differentiating this, we get

$$\frac{C}{v}dA + \frac{A}{v}dC - \frac{AC}{v^2}dv = 0$$

Dividing this equation by $\frac{A.C}{v}$, we get

$$\frac{dA}{A} + \frac{dC}{C} - \frac{dv}{v} = 0$$

The flow is adiabatic and frictionless

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$$\therefore pv^n = c \text{ (constant)}$$

$$\therefore \log_e p + n \log_e v = \log_e c$$

Differentiating, we get

$$\frac{dp}{p} + n\frac{dv}{v} = 0$$

$$\therefore \frac{dv}{v} = -\frac{1}{n}\frac{dp}{p} \qquad ...(5.16)$$

From the momentum equation for adiabatic and frictionless flow

$$CdC = -vdp$$

$$\therefore \frac{dC}{C} = -\frac{vdp}{C^2} \qquad ...(5.17)$$

Substituting values of $\frac{dC}{C}$ and $\frac{dv}{v}$ into the equation (5.15), we get

$$\frac{dA}{A} - v\frac{dp}{C^2} + \frac{1}{n}\frac{dp}{p} = 0$$

$$\therefore \frac{dA}{A} = \frac{1}{n}\frac{dp}{p}\left(\frac{npv}{C^2} - 1\right)$$

The sonic velocity of steam at pressure p and specific volume v is given by

$$a = \sqrt{n \cdot p \cdot v}, \quad \therefore a^2 = n \cdot p \cdot v$$

$$\therefore \boxed{\frac{dA}{A} = \frac{1}{n}\frac{dp}{p}\left(\frac{a^2}{C^2} - 1\right)} \qquad ...(5.18)$$

The ratio of the velocity of fluid (steam) C to the local acoustic (sound) velocity a is called the ***Mach number*** *and is denoted by the symbol M.*

$$\frac{dA}{A} = \frac{1}{n}\frac{dp}{p}\left(\frac{1}{M^2} - 1\right)$$

$$\therefore \boxed{\frac{dA}{A} = \frac{1}{n}\frac{dp}{p}\left(\frac{1 - M^2}{M^2}\right)} \qquad ...(5.19)$$

From the equation (5.19), the following points can be drawn.

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(1) Accelerated flow, $\frac{dp}{p}$ negative (flow through nozzles) :

(a) When C is less than a i.e. $M < 1$, the $\frac{dA}{A}$ must be negative to balance the equation. This means that, with decrease of area velocity increases and pressure decreases. This gives convergent part of the nozzle.

(b) When $C = a$, i.e., $M = 1$, then $\frac{dA}{A} = 0$. This means that, area is constant, no change in velocity and pressure. This corresponds to the throat of the nozzle.

(c) When C is greater than a i.e. $M > 1$, then $\frac{dA}{A}$ must be positive. This means that, with increase of area velocity increases and pressure decreases. This gives the divergent part of the nozzle.

(2) Retarded flow, $\frac{dp}{p}$ positive (flow through diffuser) :

This applies to diffusers in which the kinetic energy of flow is converted into pressure energy.

(a) When C is less than a i.e. $M < 1$, the $\frac{dA}{A}$ must be positive to balance the equation. This means that, with increase of area velocity decreases and pressure increases. This gives divergent type diffuser.

(b) When C is equal to a, i.e. $M = 1$, $\frac{dA}{A} = 0$. This means that, area is constant, no change in velocity and pressure. It corresponds to throat.

(c) When C is greater than a i.e. $M > 1$, $\frac{dA}{A}$ must be negative to balance the equation. This means that, with decrease in area, velocity decreases and pressure increases. This gives convergent type diffuser.

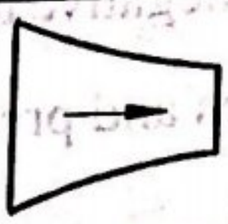
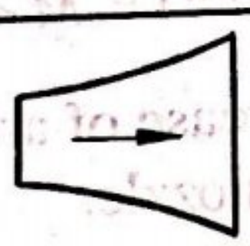
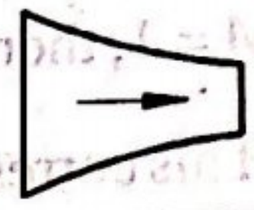
The above cases are summarized in Table 5.1.

Nozzles are used for increasing the velocity of steam. At the exit of the boiler, velocity is negligible (i.e. $M << 1$) while the outlet velocity from nozzle has to be greater than the sonic velocity (i.e. $M >> 1$). Steam first passes through subsonic nozzle and its velocity increases to sonic velocity ($M = 1$) and it reaches throat. We still want higher velocity, hence divergent nozzle has to be used. Hence the shape of the nozzle will be a combination of convergent part, a throat and a divergent part, i.e. a convergent-divergent nozzle.

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Table 1.1

Types of flow	*Accelerated flow, falling pressure*	*Retarded flow, rising pressure*
Subsonic	 *Convergent nozzle*	 *Divergent diffuser*
Supersonic	 *Divergent nozzle*	 *Convergent diffuser*

If the steam velocity at inlet to nozzle is local sonic velocity (i.e. M = 1), divergent type of nozzle will be used. If it is required that the outlet velocity from the nozzle should be only equal to sonic velocity, convergent nozzle only will be used.

***Problem** 5.7 A steam turbine develops 130 kW with a steam consumption of 12 kg/kWh. The throat diameter of nozzles is 0.7 cm each. The upstream pressure is 15 bar and temperature is 300°C. The back pressure is 0.1 bar. Neglecting velocity of approach, determine the numbers of nozzles and the steam consumption based on these conditions. Assume that the flow is isentropic between inlet and exit.*

If 10% of the isentropic heat drop is wasted between throat and exit, find the exit diameter and final condition. of steam. What will be the overall efficiency of the nozzle.

Solution : Given data (Refer Fig. 5.12)

$P = 130$ kW, $m_s = 12$ kg/kWh, $D_2 = 0.7$ cm, $p_1 = 15$ bar, $T_1 = 300°C$, $p_3 = 0.1$ bar, $C_1 = 0$, Number of nozzle = ?, m = ?, $D_3' = ?$, $x_3' = ?$, $\eta_n = ?$

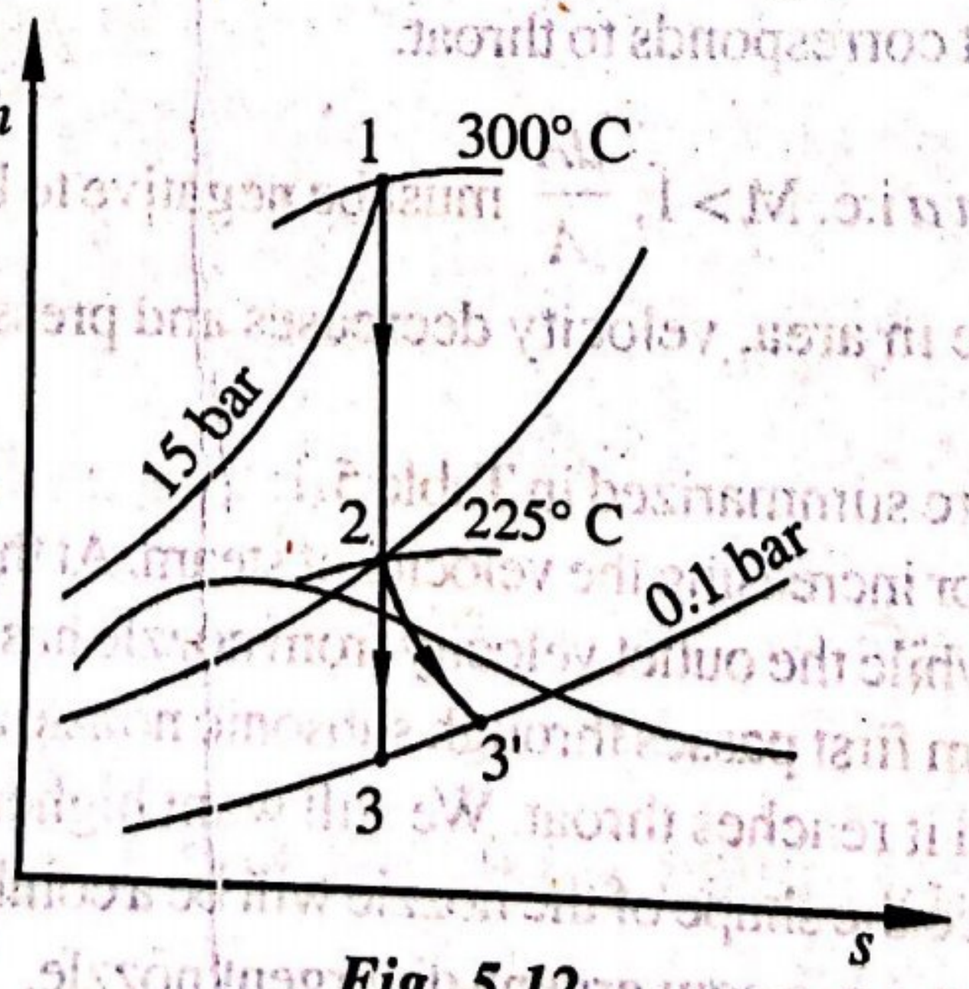


Fig. 5.12

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Let 1, 2, and 3 represent the inlet, throat and exit condition of nozzle. 3′ is the actual condition of steam at exit.

For isentropic expansion, the throat pressure is given by

$$p_2 = 0.5457\, p_1 = 0.5457 \times 15 = 8.185 \text{ bar}$$

From Mollier's diagram and steam tables

$$h_1 = 3040 \text{ kJ/kg},\ h_2 = 2890 \text{ kJ/kg},\ T_2 = 225°\text{C}$$

$$h_3 = 2190 \text{ kJ/kg},\ v_2 = 0.28 \text{ m}^3/\text{kg},$$

$$x_3 = 0.835,\ v_{g_3} = 14.675 \text{ m}^3/\text{kg}$$

$$v_3 = x_3 v_{g_3} = 0.835 \times 14.675 = 12.254\, m^3/kg$$

The velocity at the throat is given by

$$C_2 = 44.72\sqrt{h_1 - h_2} = 44.72\sqrt{3040 - 2890} = 547.706 \text{ m/s}$$

Throat area, $A_2 = \frac{\pi}{4} D_2^2 = \frac{\pi}{4} \times (0.7 \times 10^{-2})^2 = 3.848 \times 10^{-5}\, m^2$

$$\text{Steam consumption} = \frac{12 \times 130}{3600} = 0.4333\, kg/s$$

The mass flow rate per nozzle is given by using the continuity equation.

$$m = \frac{A_2 \times C_2}{v_2} = \frac{3.848 \times 10^{-5} \times 547.706}{0.28} = 0.0753\, kg/s$$

$$\therefore \text{Number of nozzles} = \frac{\text{Steam consumption (Total discharge)}}{\text{Discharge through one nozzle}}$$

$$= \frac{0.4333}{0.0753} = 5.75 \text{ say } \mathbf{6} \quad \textbf{Ans}$$

$$\text{Actual steam consumption} = 6 \times 0.0753 = 0.4518 \text{ kg/s}$$

$$= \frac{0.4518 \times 3600}{130} = \mathbf{12.511 \text{ kg/kWh}} \quad \textbf{Ans}$$

If 10% of isentropic heat drop is wasted between throat and exit, then

$$h_1 - h_3' = 0.9(h_2 - h_3) + (h_1 - h_2)$$

$$3040 - h_3' = 0.9(2890 - 2190) + (3040 - 2890)$$

$$\therefore h_3' = 2260 \text{ kJ/kg}$$

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From the Mollier's diagram, $x_3' = 0.85$

Actual velocity at the exit is given by

$$C_3' = 44.72\sqrt{h_1 - h_3'} = 44.72\sqrt{3040 - 2260} = 1248.96\, m/s$$

$$v_3' = x_3' \times v_{g_3} = 0.85 \times 14.675 = 12.474\ m^3/kg$$

Area at exit is given by continuity equation

$$A_3' = \frac{mv_3'}{C_3'} = \frac{0.0753 \times 12.474}{1248.96} = 7.52 \times 10^{-4} m^2$$

$$A_3' = \frac{\pi}{4} D_3'^2 = 7.52 \times 10^{-4} = \frac{\pi}{4} \times D_3'^2$$

$$\therefore D_3' = 0.03094\, m = \mathbf{3.094\ cm} \quad \textbf{Ans}$$

Overall efficiency of nozzle is given by

$$\eta_n = \frac{h_1 - h_3'}{h_1 - h_3} = \frac{3040 - 2260}{3040 - 2190} = 0.9176 = \mathbf{91.76\%} \quad \textbf{Ans}$$

Problem 5.8 *A convergent-divergent nozzle is required to discharge 3 kg of steam per second. The nozzle is supplied with steam at 8 bar and 200°C and discharge takes place against a back pressure of 1 bar. The expansion upto throat is isentropic and the frictional resistance between the throat and exit is equivalent to 70 kJ/kg of steam. Taking approach velocity of 70 m/s and throat pressure of 4 bar, determine : (i) suitable areas for the throat and exit, and (ii) overall efficiency of the nozzle based on the enthalpy drop between the actual inlet pressure and temperature and the exit pressure.*

Solution : Given data : (Refer Fig. 5.13)

$m = 3$ kg/s, $p_1 = 8$ bar, $T_1 = 200°C$, $p_3 = 1$ bar, Frictional resistance = 70 kJ/kg of steam, $C_1 = 70$ m/s, $p_2 = 4$ bar, $A_2 = ?$, $A_3' = ?$, $\eta_n = ?$

(i) Areas for the throat and exit :

The expansion of steam through the nozzle on the Mollier diagram is shown in Fig. 5.13. From the Mollier diagram,

$h_1 = 2840$ kJ/kg, $h_2 = 2700$ kJ/kg, $x_2 = 0.985$, $h_3 = 2470$ kJ/kg, $x_3 = 0.91$

From steam tables, the specific volume of steam at throat corresponding to 4 bar

$$v_{g_2} = 0.462\ m^3/kg$$

$$\therefore v_2 = x_2 v_{g_2} = 0.985 \times 0.462 = 0.45507\ m^3/kg$$

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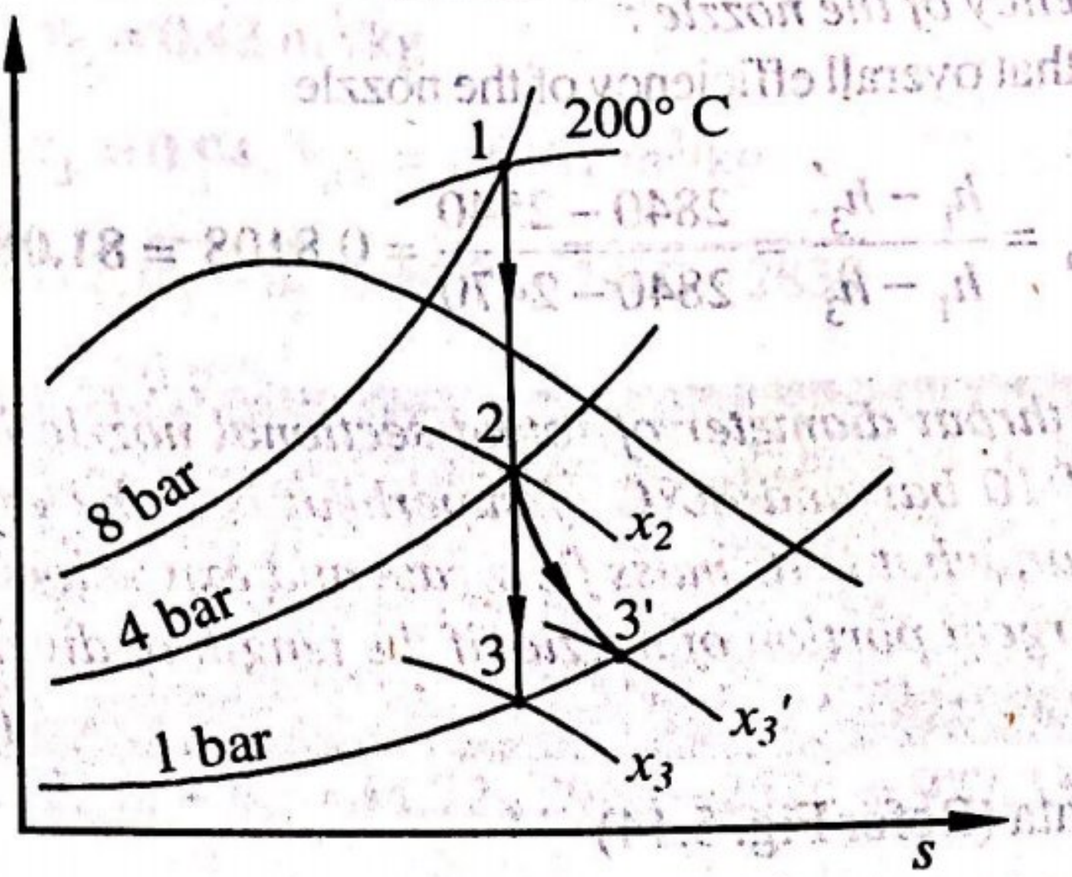


Fig. 5.13

Velocity of steam at throat is given by

$$C_2 = \sqrt{2(h_1 - h_2)\times 1000 + C_1^2}$$

$$= \sqrt{2(2840 - 2700)\times 1000 + 70^2} = 533.76\, m/s$$

Area at throat is given by continuity equation

$$A_2 = \frac{mv_2}{C_2} = \frac{3\times 0.45507}{533.76} = \mathbf{2.558 \times 10^{-3}\ m^2} \quad \textit{Ans}$$

Since there is a frictional resistance of 70 kJ/kg of steam between the throat and exit therefore,

$$h_3' - h_3 = 70 \quad \therefore h_3' = h_3 + 70 = 2470 + 70 = 2540 \text{ kJ/kg}$$

$\therefore x_3' = 0.94$ (from Mollier-diagram)

Specific volume of steam corresponding to 1 bar, $v_{g_3} = 1.694\ m^3/kg$

$$\therefore v_3' = x_3' v_{g_3} = 0.94\times 1.694 = 1.592\ m^3/kg$$

Velocity of steam at exit is given by

$$C_3' = \sqrt{2(h_1 - h_3')\times 1000 + C_1^2}$$

$$= \sqrt{2(2840 - 2540)\times 1000 + 70^2} = 777.75 \text{ m/s}$$

Area at exit is given by continuity equation

$$A_3' = \frac{mv_3'}{C_3'} = \frac{3\times 1.592}{777.75} = \mathbf{6.1408 \times 10^{-3}\ m^2} \quad \textit{Ans}$$

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(ii) Overall efficiency of the nozzle :

We know that overall efficiency of the nozzle

$$\eta_n = \frac{h_1 - h_3'}{h_1 - h_3} = \frac{2840 - 2540}{2840 - 2470} = 0.8108 = \mathbf{81.08\%}$$ **Ans**

Problem 5.9 *The throat diameter of round sectional nozzle is 0.6 cm. Steam with an initial pressure of 10 bar and 80°C of superheat is to be expanded down to a back pressure of 1.05 bar, what is its mass flow rate and exit velocity ? What is the angle of cone used for divergent portion of nozzle, if the length of divergent portion is 5 cm ?* **[G.U. Dec. '04, 8 Marks]**

Solution : Given data (Refer Fig. 5.14)

$p_1 = 10$ bar, Degree of superheat = 80°C, $p_3 = 1.05$ bar, $D_2 = 0.6$ cm, $m = ?$,

$C_3 = ?$, length of divergent portion = 5 cm, $2\theta = ?$

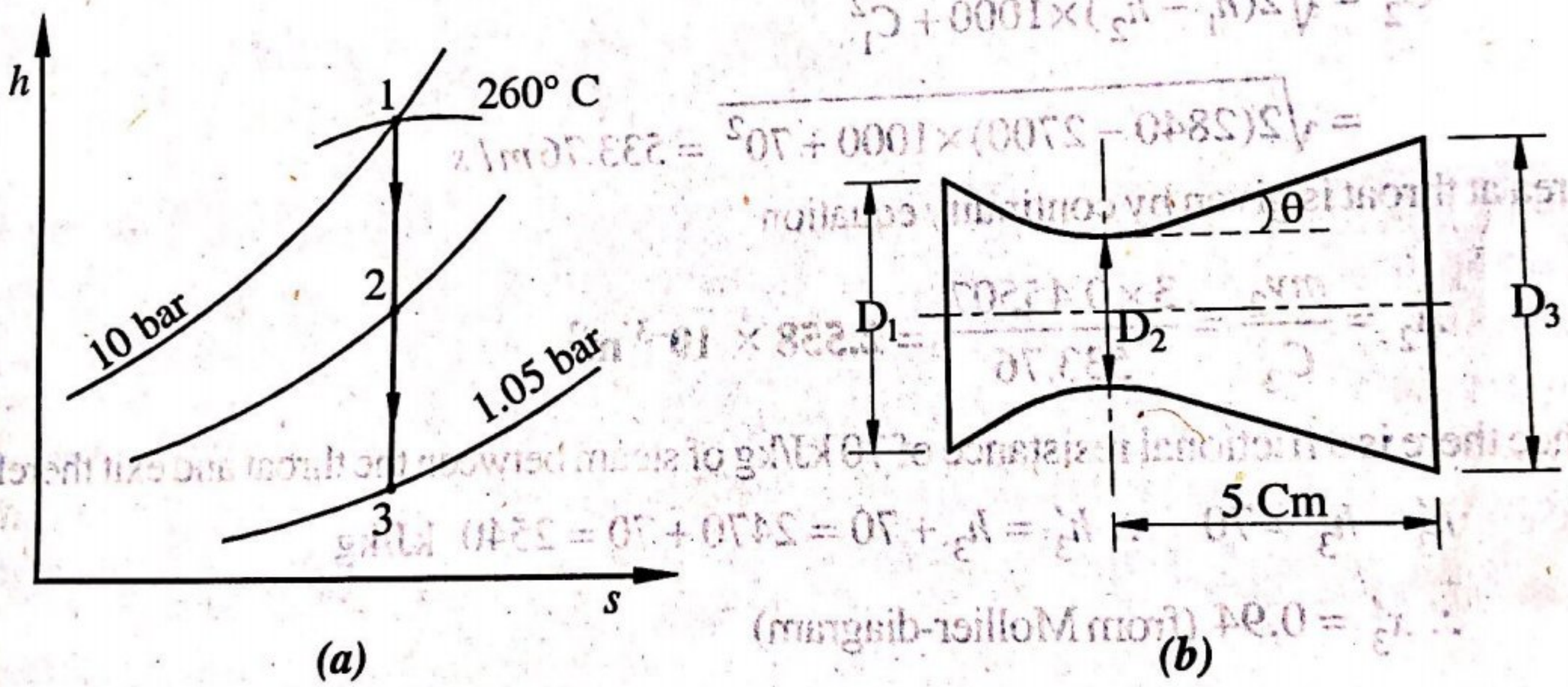


Fig. 5.14

From steam tables, at 10 bar pressure

$$T_{sat} = 179.9^\circ C$$

We know that Degree of superheat = $T_{sup} - T_{sat}$

$$80 = T_{sup} - 179.9$$

$$\therefore T_1 = T_{sup} = 259.9^\circ C$$

From Mollier diagram, $h_1 = 2960$ kJ/kg

The throat pressure is given by

$$p_2 = p_1 \times 0.5457 = 10 \times 0.5457 = 5.457 \text{ bar}$$

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$\therefore\ h_2 = 2830$ kJ/kg, $v_2 = 0.43$ m^3/kg

$\therefore\ h_3 = 2530$ kJ/kg, $x_3 = 0.94$, $v_{g3} = 1.6181$ m^3/kg

Velocity at throat, $C_2 = 44.72\sqrt{h_1 - h_2} = 44.72\sqrt{2960 - 2830}$

$= 509.886$ m/s

The mass flow rate, $m = \dfrac{A_2 C_2}{v_2} = \dfrac{\frac{\pi}{4}(0.6\times10^{-2})^2 \times 509.886}{0.43}$

$m = \mathbf{0.03353}$ **kg/s** ***Ans***

Velocity at exit, $C_3 = 44.72\sqrt{h_1 - h_3} = 44.72\sqrt{2960 - 2530} = \mathbf{927.3336}$ **m/s** ***Ans***

Area at exit, $A_3 = \dfrac{mv_3}{C_3} = \dfrac{m \times x_3 v_{g_3}}{C_3} = \dfrac{0.03353\times0.94\times1.6181}{927.3336}$

$A_3 = 5.5\times10^{-5}\, m^2 = \dfrac{\pi}{4} D_3^2$

$\therefore\ D_3 = 8.368\times10^{-3}\, m = 0.8368\ cm$

From the Fig. 5.14(b)

$$\tan\theta = \frac{D_3 - D_2}{2L} = \frac{0.8368 - 0.6}{2\times5} = 0.02368$$

$\therefore$ Semi-cone angle, $\theta = 1.356$ $\therefore\ 2\theta = \mathbf{2.712}$ ***Ans***

Problem 5.10 *A nozzle is supplied with steam at 8 bar and 250°C, expands to 1 bar isentropically. If the divergent part of nozzle is 5 cm long and throat diameter is 0.75 cm, determine the angle of cone required for divergent part of nozzle. Considering 15% loss of heat drop occurs in the divergent part only.* ***[G.U., Dec. 06, 8 Marks]***

Solution : Given data : (Refer Fig. 5.15)

$p_1 = 8$ bar, $T_1 = 250$°C, $p_3 = 1$ bar, L = 5 cm, $D_2 = 0.75$ cm, $\eta_n = (1–0.15) = 0.85$,

$2\theta = ?$

The throat pressure is given by

$$p_2 = p_1 \times 0.5457 = 8\times0.5457 = 4.366 \text{ bar}$$

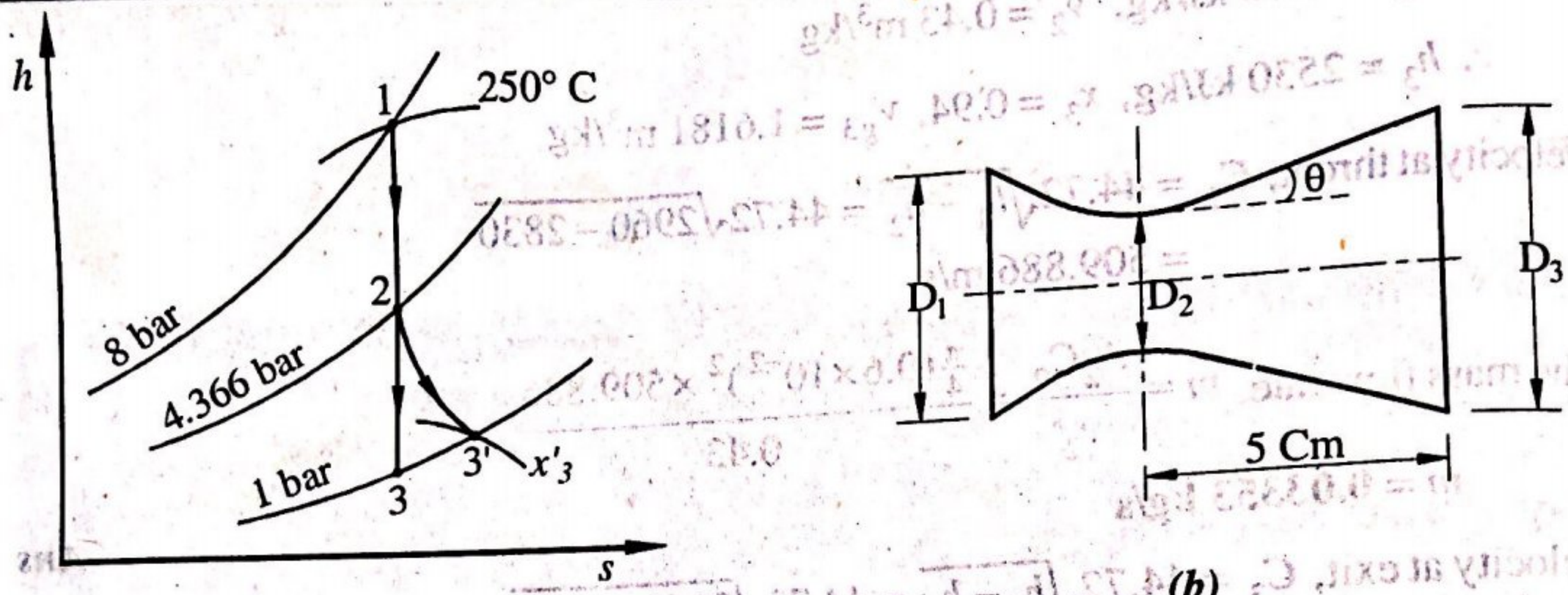
From Mollier diagram

$h_1 = 2950$ kJ/kg, $h_2 = 2810$ kJ/kg , $v_2 = 0.49$ m^3/kg

$h_3 = 2560$ kJ/kg, $v_{g_3} = 1.694$ m^3/kg

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(a) (b)

Fig. 5.15

Velocity at throat, $C_2 = 44.72\sqrt{h_1 - h_2} = 44.72\sqrt{2950 - 2810}$

$= 529.134$ m/s

Nozzle efficiency, $\eta_n = \dfrac{h_2 - h_3'}{h_2 - h_3}$

$\therefore h_3' = h_2 - \eta_n (h_2 - h_3)$

$= 2810 - 0.85(2810 - 2560) = 2597.5$ kJ/kg

From Mollier diagram, $x_3' = 0.965$

Specific volume of steam at exit, $v_3' = x_3' v_{g_3} = 0.965 \times 1.694 = 1.6347\ m^3/kg$

The mass flow rate, $m = \dfrac{A_2 C_2}{v_2}$

$= \dfrac{\frac{\pi}{4} \times (0.75 \times 10^{-2})^2 \times 529.134}{0.49}$

$= 0.04771$ kg/s

Velocity at exit, $C_3' = 44.72\sqrt{h_1 - h_3'} = 44.72\sqrt{2950 - 2597.5}$

$= 839.617$ m/s

Actual area at exit, $A_3' = \dfrac{m v_3'}{C_3'} = \dfrac{0.04771 \times 1.6357}{839.617}$

$= 9.289 \times 10^{-5}\ m^2$

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$$\therefore \frac{\pi}{4} D_3'^2 = 9.289 \times 10^{-5}$$

$$\therefore D_3' = 0.010875\, m = 1.0875\, cm$$

From Fig. 5.15(b)

$$\tan\theta = \frac{D_3' - D_2}{2L} = \frac{1.0875 - 0.75}{2 \times 5} = 0.03375$$

$\therefore$ Semi-cone angle, $\theta = 1.933^\circ$

$\therefore 2\theta = \mathbf{3.866^\circ}$ ***Ans***

Problem 5.11 *The nozzle of a De Laval steam turbine are supplied with dry saturated steam at a pressure of 10 bar. The pressure at the outlet is 1 bar. The turbine has four nozzles with a throat diameter of 2 mm. Assuming nozzle efficiency as 89% and that of turbine rotor 70%, find the quantity of steam used per hour and the power developed.*

Solution : Given data : (Refer Fig. 5.16)

$p_1 = 10$ bar, $p_3 = 1$ bar, Number of nozzles = 4, $D_2 = 2$ mm, $\eta_n = 89\%$, $\eta_t = 70\%$,

$m = ?$, $P = ?$

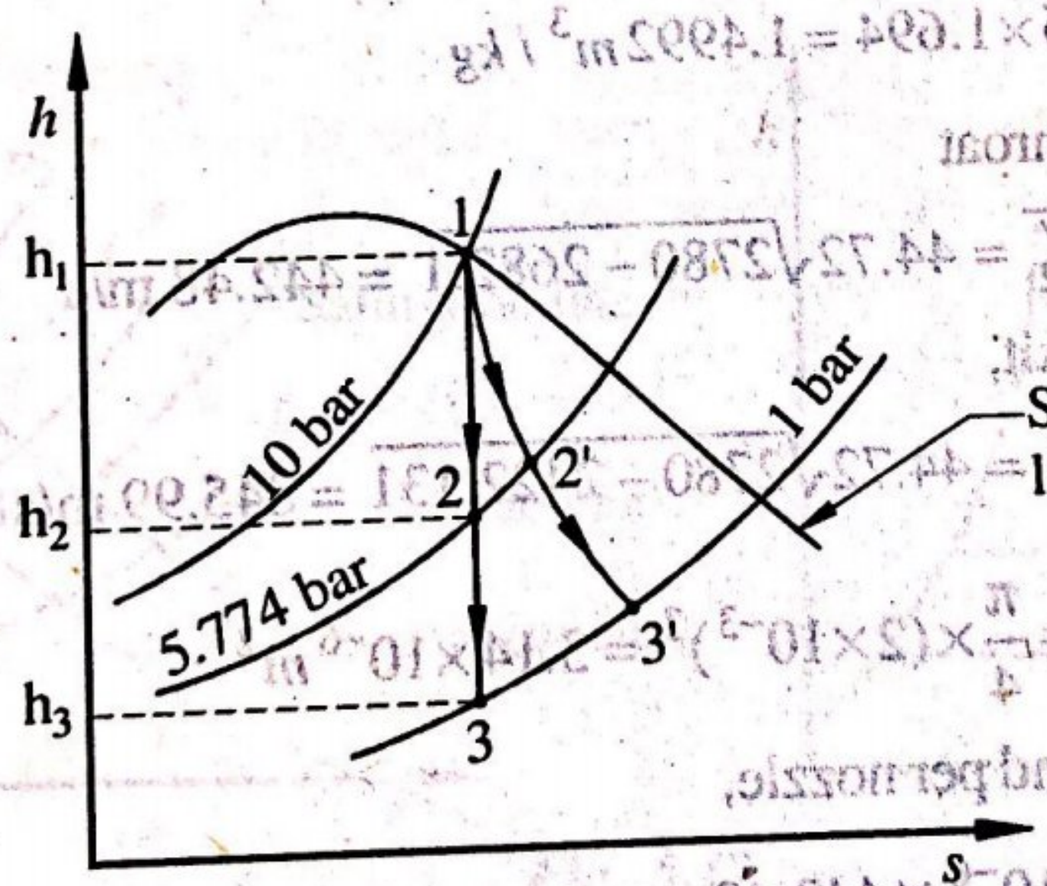


Fig. 5.16

We know that, $\frac{p_2}{p_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$

Since the steam is dry and saturated, $n = 1.135$

$$\therefore \frac{p_2}{p_1} = \left(\frac{2}{1.135+1}\right)^{\frac{1.135}{1.135-1}} = 0.5774$$

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$$\therefore\ p_2 = 0.5774\, p_1 = 0.5774 \times 10 = 5.774 \text{ bar}$$

From Mollier chart

$$h_1 = 2780 \text{ kJ/kg},\ h_2 = 2670 \text{ kJ/kg},\ h_3 = 2390 \text{ kJ/kg}$$

From steam tables :

$$v_{g_2} = 0.3257 \text{ m}^3/\text{kg (at 5.8 bar)},\ v_{g_3} = 1.694 \text{ m}^3/\text{kg (at 1 bar)}$$

Enthalpy at point $2'$,

$$h_2' = h_1 - \eta_n (h_1 - h_2) = 2780 - 0.89(2780 - 2670)$$
$$= 2682.1 \text{ kJ/kg}$$

Enthalpy at point $3'$

$$h_3' = h_2' - \eta_n (h_2' - h_3) = 2682.1 - 0.89(2682.1 - 2390)$$
$$= 2422.131 \text{ kJ/kg}$$

From Mollier chart

$$x_2' = 0.97,\ x_3' = 0.885$$

$$v_2' = x_2' v_{g_2} = 0.97 \times 0.3257 = 0.31593\, m^3/kg$$

$$v_3' = x_3' v_{g_3} = 0.885 \times 1.694 = 1.4992\, m^3/kg$$

Actual velocity of steam at throat

$$C_2' = 44.72\sqrt{h_1 - h_2'} = 44.72\sqrt{2780 - 2682.1} = 442.48 \text{ m/s}$$

Actual velocity of steam at exit,

$$C_3' = 44.72\sqrt{h_1 - h_3'} = 44.72\sqrt{2780 - 2422.131} = 845.99 \text{ m/s}$$

Area at throat, $A_2 = \frac{\pi}{4} D_2^2 = \frac{\pi}{4} \times (2 \times 10^{-3})^2 = 3.14 \times 10^{-6}\, m^2$

Mass of steam used per second per nozzle,

$$m = \frac{A_2 C_2'}{v_2'} = \frac{3.14 \times 10^{-6} \times 442.48}{0.31593} = 4.3978 \times 10^{-3}\, kg/s$$

Total mass of steam used per second, m_t = Number of nozzle $\times\ m$

$$= 4 \times 4.3978 \times 10^{-3} = 0.01759 \text{ kg/s}$$

$\therefore$ Quantity of steam used per hour

$$= 0.01759 \times 3600 = \mathbf{63.324 \text{ kg/h}}$$

Energy supplied by the steam to the wheel per second **Ans**

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$$= \frac{m_t C_3'^2}{2} = \frac{0.01759 \times (845.99)^2}{2}$$

$$= 6294.57 \text{ W} \cong 6.2945 \text{ kW}$$

$\therefore$ Power developed, $P = \eta_t \times 6.2945 = 0.7 \times 6.2945 = \mathbf{4.406\ kW}$ ***Ans***

5.9 Super saturated flow in nozzles (Metastable flow) [May '15, May '13]

When dry and saturated steam is caused to expand in a nozzle, the actual measured steam flow is found to be greater than the theoretically calculated discharge. This phenomenon is the reverse of what would naturally be expected. In view of the friction losses which occur in nozzle, it would be expected that the actual discharge would be less than the theoretical value. Thus we would expect the actual mass flow to be about 3 to 5 percent less than the theoretical, whereas experiments have shown it to be from 1 to 3 percent greater. On the other hand, the actual discharge of superheated steam is in good agreement with the theoretical value when allowance is made for friction losses. *The increase in measured discharge to the theoretically calculated discharge is due to the time lag in the condensation of steam and thus the steam remains in dry state instead of wet; this phenomenon is known as* ***supersaturation***. Sometimes the steam in this condition is said to be in a ***metastable state.*** This has been investigated by Wilson and Martin.

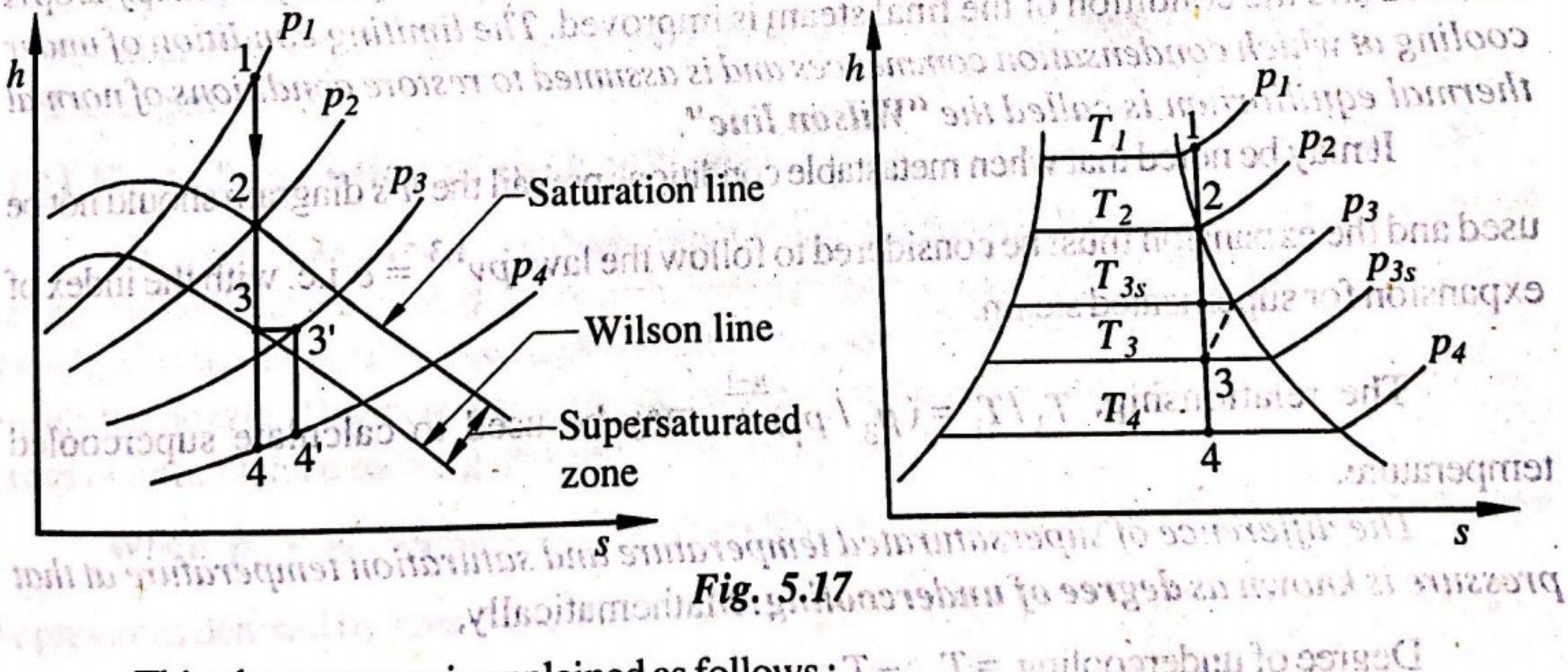


Fig. 5.17

This phenomenon is explained as follows :

The converging part of the nozzle is so short and the steam velocity so high that the molecules of steam have insufficient time to collect and form droplets so that normal condensation does not take place. Such rapid expansion is said to be metastable and produces a supersaturated state. In this state of supersaturation *the steam is undercooled to a temperature less than that corresponding to its pressure; consequently the density of steam increases and hence the mass flow rate*. Prof. Wilson through experiments showed that dry saturated steam, when suddenly expanded in the absence of dust, does not condense until its density is

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about 8 times that of the saturated vapour of the same pressure. This effect is discussed below:

The supersaturation phenomenon is shown on the *h-s* and *T-s* diagram in the Fig. 5.17. The point 1 represents initial state of the steam in the superheated region on the pressure line p_1. 1 to 2 shows the isentropic expansion in thermal equilibrium upto saturation. In natural course at the point 2 where the pressure is p_2, a change of phase must begin to occur i.e. condensation should start at point 2, but due to supersaturation the achievement of the equilibrium between the liquid and vapour phase is delayed and the vapour continues to expand in a dry state instead of wet state. Thus, steam expands isentropically without condensation to point 3, 3 being on the superheat constant pressure curve p_3 produced. At point 3, the limit of supersaturation is reached and steam reverts to its normal condition at 3′ at the same enthalpy value as 3 and at the same pressure. The process 2 to 3 shows the expansion under supersaturation condition which is not in thermal equilibrium. *The zone between the Wilson line and dry saturated line is called the* ***supersaturated zone*** *and the flow through this zone is called the* ***supersaturated flow***. This is also called ***undercooled*** because at any pressure between p_2 and p_3. i.e. within the super saturated zone, the temperature of the vapour is always less than saturation temperature corresponding to that pressure. The steam continuous expanding isentropically to a lower pressure to point 4′ instead of 4 which would have been reached if thermal equilibrium had been maintained. Consequently, enthalpy drop is reduced and the condition of the final steam is improved. *The limiting condition of under cooling at which condensation commences and is assumed to restore conditions of normal thermal equilibrium is called the* ***"Wilson line"***.

It may be noted that when metastable conditions prevail the *h-s* diagram should not be used and the expansion must be considered to follow the law $pv^{1.3} = c$ i.e. with the index of expansion for superheated steam.

The relationship, $T_3 / T_1 = (p_3 / p_1)^{\frac{n-1}{n}}$ may be used to calculate supercooled temperature.

The difference of supersaturated temperature and saturation temperature at that pressure is known as ***degree of undercooling***. Mathematically,

$$\text{Degree of undercooling} = T_{3s} - T_3$$

The ratio of pressure corresponding to temperature of super saturated and pressure corresponding to saturation temperature is known as ***degree of supersaturation***. Mathematically,

$$\text{Degree of supersaturation} = \frac{p_3}{p_{3s}}$$

The problems on supersaturated flow cannot be solved by Mollier diagram unless

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Wilson line is drawn on it.

The velocity of steam at the end of expansion is found by using the relation

$$C_2 = \sqrt{\frac{2n}{n-1} p_1 v_1 \left(1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right)}$$

where subscripts 1 and 2 refer to entry and throat conditions

$$\text{Enthalpy drop} = \frac{n}{n-1} p_1 v_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

- **Effects of supersaturation :**

The following are the effects of supersaturation flow in a nozzle :

(1) The supersaturation reduces the heat drop for the same pressure limits, below that for thermal equilibrium. Hence the exit velocity of the steam is reduced.

(2) The supersaturation increases dryness fraction of steam.

(3) The supersaturation increases the entropy and specific volume of the steam.

(4) Since the condensation does not take place during supersaturated expansion, so the temperature, at which the supersaturation occurs will be less than the saturation temperature corresponding to the pressure. Therefore, the density of supersaturated steam will be more than that for the equilibrium conditions which gives the increase in the mass of steam discharged.

5.10 Effect of variation of back pressure [May '15]

Let us consider a convergent-divergent nozzle as shown in Fig. 5.18, which also show the effect of variation of back pressure in a convergent-divergent nozzle. The inlet conditions of the fluid is assumed to be constant but back pressure is varied by means of a valve placed outside the nozzle. The pressure along the longitudinal section of the nozzle is measured by means of a search tube running centrally along its length.

When $p_1 = p_b$, there is no flow through nozzle. When back pressure is lowered to the pressures denoted by lines 6 and 5, so that $p_b / p_1 < 1$, but $p_b / p_1 > p_c / p_1$, the velocity increases in the convergent section, but M < 1 occurs at the throat. The divergent section acts as the subsonic diffuser in which pressure increases and velocity decreases.

The ideal throat pressure ($p_c = 0.5774\ p_1$) is shown at C where the velocity is sonic and we see that the pressure of curves 2, 3 and 4 after the throat are well below the ideal throat pressure, so that the velocities beyond the throat are supersonic.

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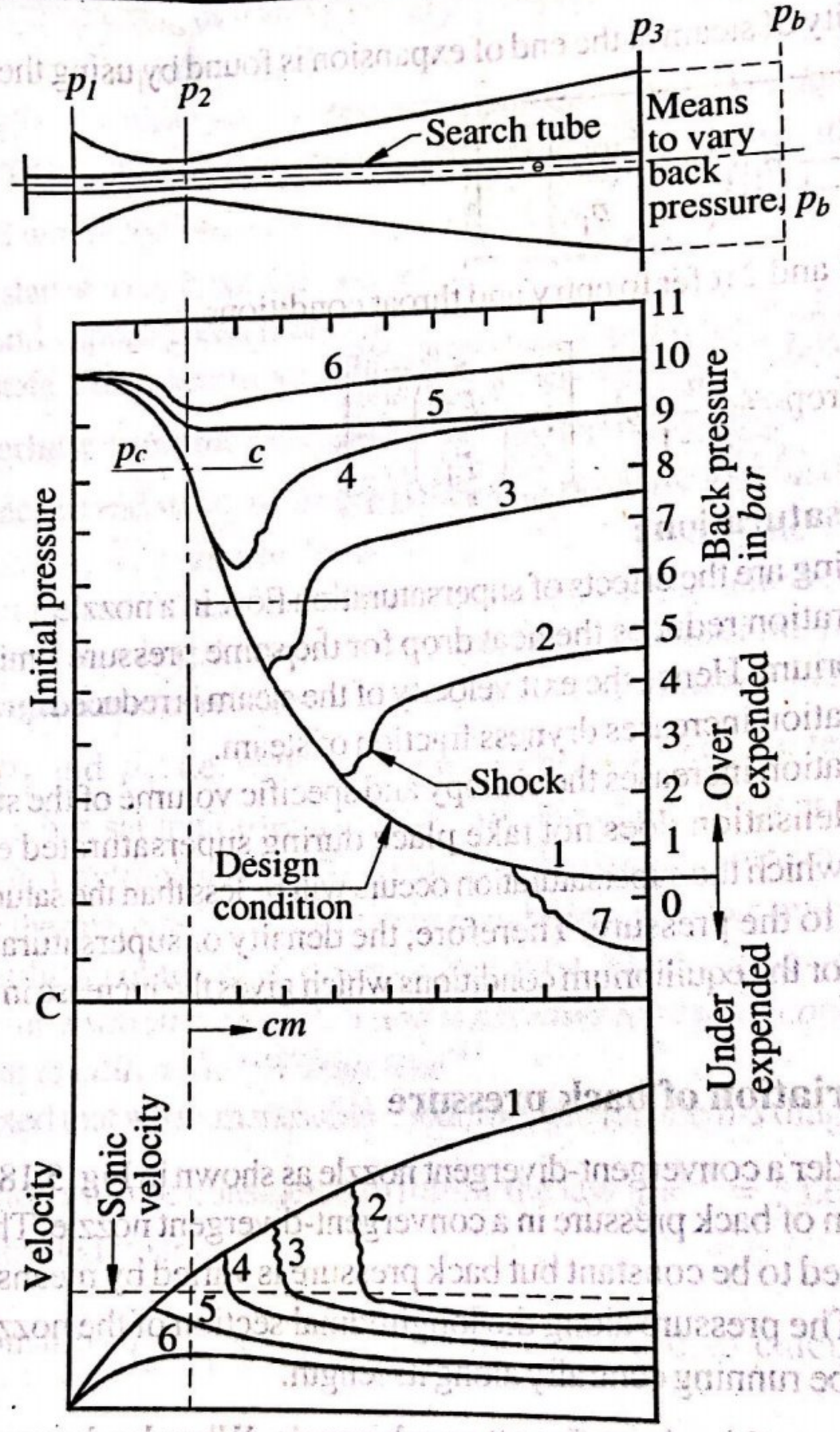


Fig. 5.18 Effect of variation of back pressure

The curve 1 indicates back pressure for which the flow is isentropic throughout and the diverging section acts as a supersonic nozzle with a continuous decrease in pressure and a continuous increase in velocity, and $p_3 = p_b$. This condition of supersonic flow indicates the design pressure ratio of the nozzle. *When the back pressure of a nozzle is below the design value the nozzle is said to* ***underexpand***. In such case the fluid expands to the design pressure in the nozzle and then expands violently and irreversibly down to the back pressure on leaving the nozzle (curve 7). *If the back pressure of a nozzle is above the design value, the nozzle is said to* ***overexpand***.

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The back pressure between p_c and indicated by curve 1, flow is not isentropic in the diverging part, and it is accompanied by an irreversible phenomenon known as shocks. Shocks occur only when the flow is supersonic, and after the shock the flow becomes subsonic and the rest of the duct acts as a diffuser. Properties vary discontinuously across the shock. When the back pressure is as indicated by curve 2, the flow through the nozzle is isentropic, with pressure continuously decreasing the velocity increasing, but a shock appears just before the exit of the nozzle. When the back pressure is increased from 2 to 3, the shock moves upstream as shown in Fig. 5.18. When the back pressure is further increased, the shock moves upstream as indicated by curve 4 and disappears at the nozzle throat where the back pressure equal to p_c. In this case, shock does not occur because supersonic velocity is not achieved.

Problem 5.12 *The dry saturated steam is expanded in a nozzle from pressure of 8 bar to a pressure of 4 bar. If the expansion is supersaturated, find (i) the degree of undercooling, and (ii) the degree of supersaturation.*

Solution : $p_1 = 8$ bar, $p_2 = 4$ bar

Subscripts 1 and 2 refer to the initial and final condition of steam under this flow respectively.

(i) Degree of undercooling :

From steam tables, corresponding to a pressure of 8 bar, we find that the initial temperature of steam.

$$T_1 = 170.4^\circ C = 170.4 + 273 = 443.4 \text{ K}$$

Let T_2 = Temperature at which supersaturation occurs.

The actual temperature of steam at exit is given by

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{(n-1)}{n}}$$

In supersaturated flow, $n = 1.3$

$$\therefore T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{0.3}{1.3}} = 443.4\left(\frac{4}{8}\right)^{\frac{0.3}{1.3}}$$

$$= 377.857 \text{ K} = 104.857^\circ C$$

From steam tables, corresponding to a pressure of 4 bar, we find that the saturation temperature,

$$T_{2s} = 143.6^\circ C$$

$\therefore$ Degree of undercooling $= T_{2s} - T_2 = 143.6 - 104.857 = \mathbf{38.743^\circ C}$ ***Ans***

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(ii) Degree of supersaturation :

From steam tables corresponding to a temperature of $T_2 = 104.857^\circ C$, we find that

$$p_{2s} = 1.2 \text{ bar}$$

$$\therefore \text{Degree of supersaturation} = \frac{p_2}{p_{2s}} = \frac{4}{1.2} = 3.333 \quad \textit{Ans}$$

Problem 5.13 *Dry saturated steam at 4 bar passes through a convergent-divergent nozzle of throat area 30 cm² to a final pressure of 1 bar. Determine the mass of steam discharged per hour if (i) the steam expands in thermal equilibrium, (ii) the steam is supersaturated during expansion. Also find the degree of undercooling in case (ii).*

Solution : Given data

$p_1 = 4$ bar, $p_3 = 1$ bar, $A_2 = 30 \text{ cm}^2$, $m = ?$, Degree of undercooling = ?

(i) The steam is expanded in thermal equilibrium :

For isentropic expansion, the mass flow rate is given by

$$\frac{m}{A_2} = \sqrt{\frac{2n}{n-1}\frac{p_1}{v_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right]}$$

For dry and saturated steam,

$$n = 1.135, \; p_2 = 0.5774\, p_1$$

$$\therefore \frac{m}{A_2} = 0.63599\sqrt{\frac{p_1}{v_1}}$$

From steam tables, corresponding to 4 bar pressure,

$$v_1 = 0.4622 \text{ m}^3/\text{kg}$$

$$\therefore \frac{m}{30\times10^{-4}} = 0.63599\sqrt{\frac{4\times10^5}{0.4622}}$$

$$\therefore m = 1.775\, kg/s = \mathbf{6389.826 \text{ kg/h}} \quad \textit{Ans}$$

(ii) The steam is supersaturated during expansion :

When the flow is supersaturated, $n = 1.3$, even the initial condition is dry.

The mass flow rate is given by

$$\frac{m}{A_2} = \sqrt{\frac{2n}{n-1}\frac{p_1}{v_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right]}$$

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Here, $n = 1.3,\ p_2 / p_1 = 0.5457$

$$\therefore \frac{m}{A_2} = 0.66726\sqrt{\frac{p_1}{v_1}}$$

$$\therefore \frac{m}{30\times10^{-4}} = 0.66726\sqrt{\frac{4\times10^5}{0.4622}}$$

$\therefore m = 1.862$ kg/s = **6703.998 kg/h** ***Ans***

Now, $\frac{T_3}{T_1} = \left(\frac{p_3}{p_1}\right)^{\frac{n-1}{n}}$; $T_3 = T_1\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{n}}$

From steam tables at $p_1 = 4$ bar pressure,

$$T_1 = 143.6^o C = 143.6 + 273 = 416.6\,K$$

$$\therefore T_3 = 416.6\left(\frac{1}{4}\right)^{\frac{0.3}{1.3}} = 302.54\,K$$

$\therefore T_3 = 29.54$°C

At pressure 1 bar, the saturation temperature is $T_{3s} = 99.63$°C

$\therefore$ Degree of undercooling $= T_{3s} - T_3 = 99.63 - 29.54 =$ **70.09°C** ***Ans***

Problem 5.14 *A nozzle is supplied with steam at 10 bar and 300°C, expands to 1 bar. The condensation does not take place while the steam is in the nozzle due to supersaturation phenomenon. Determine the throat area, degree of undercooling, degree of supersaturation, exit area, actual heat drop, the isentropic heat drop and increase in entropy. The mass flow rate is 1 kg/s.*

Solution : Given data (Refer Fig. 5.19)

$p_1 = 10$ bar, $T_1 = 300$°C, $p_3 = 1$ bar, $m = 1$ kg/s, $A_2 = ?$, $A_3 = ?$, $h_1 - h_3 = ?$,

$h_1 - h_{3isen} = ?$, Increase in entropy = ?, degree of undercooling, degree of supersaturation =?

1, 2, and 3 represent the inlet, throat and outlet conditions of nozzle.

$T_1 = 300 + 273 = 573$ K

Due to supersaturated flow 3 is the actual condition of steam at the exit. For supersaturated flow, index $n = 1.3$.

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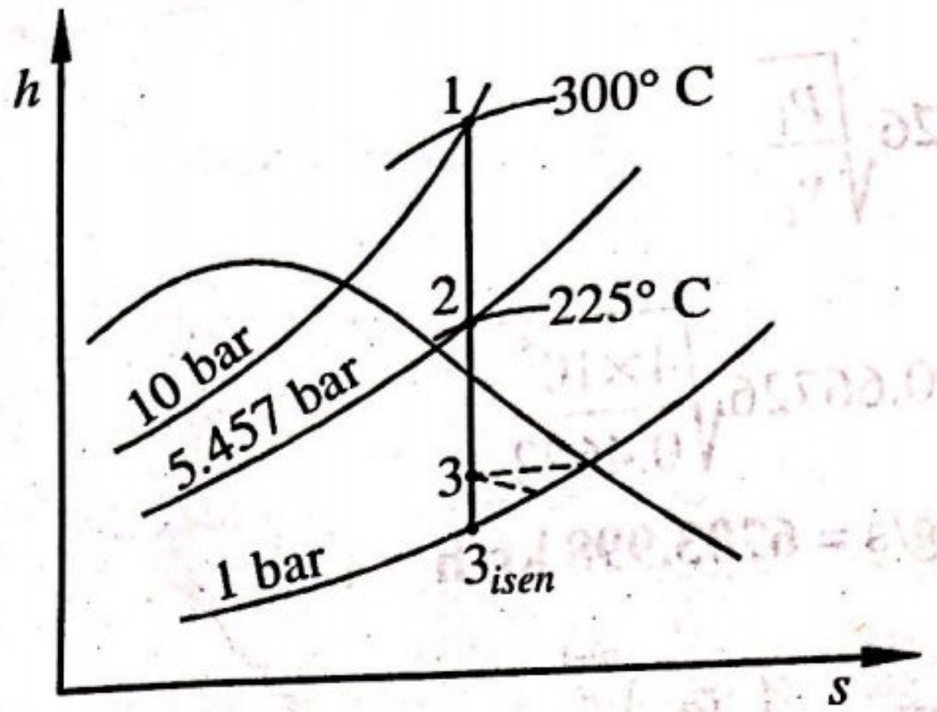


Fig. 5.19

$$\therefore \frac{T_1}{T_3} = \left(\frac{p_1}{p_3}\right)^{\frac{n-1}{n}} = \left(\frac{10}{1}\right)^{\frac{0.3}{1.3}} = 1.7012$$

$$\therefore T_3 = \frac{573}{1.7012} = 336.82\,K = 63.82^{o}\,C$$

From steam tables, corresponding to 1 bar, saturation temperature is

$$T_{3s} = 99.63°C$$

$\therefore$ Degree of undercooling $= T_{3s} - T_3 = 99.63 - 63.82 = \mathbf{35.81°C}$ ***Ans***

From steam tables, corresponding to $T_3 = 63.82°C$, saturation pressure,

$$p_{3s} = 0.24 \text{ bar}$$

$\therefore$ Degree of supersaturation $= \dfrac{p_3}{p_{3s}} = \dfrac{1}{0.24} = \mathbf{4.1667}$ ***Ans***

Throat pressure is given by

$$p_2 = p_1\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} = 10\left(\frac{2}{1.3+1}\right)^{\frac{1.3}{1.3-1}} = 5.457 \text{ bar}$$

From Mollier diagram

$$h_1 = 3050 \text{ kJ/kg},\ h_2 = 2910 \text{ kJ/kg},\ T_2 = 225°C,\ v_2 = 0.41 \text{ m}^3/\text{kg}$$

$\therefore$ Actual velocity at throat, $C_2 = 44.72\sqrt{h_1 - h_2} = 44.72\sqrt{3050 - 2910} = 529.134$ m/s

$\therefore$ Area at throat, $A_2 = \dfrac{mv_2}{c_2} = \dfrac{1 \times 0.41}{529.134} = 7.7485 \times 10^{-4}\,m^2 = \mathbf{7.7485\ cm^2}$ ***Ans***

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Enthalpy drop at the exit,

$$(h_1 - h_3) = \frac{n}{n-1} p_1 v_1 \left[1 - \left(\frac{p_3}{p_1}\right)^{\frac{n-1}{n}}\right]$$

$$= \frac{1.3}{1.3-1} \times 10 \times 10^5 \times 0.258 \left[1 - \left(\frac{1}{10}\right)^{\frac{0.3}{1.3}}\right]$$

$$= 460839.6 \text{ J/kg} \cong \mathbf{460.84 \text{ kJ/kg}} \quad \textit{Ans}$$

The exit velocity

$$C_3 = 44.72\sqrt{h_1 - h_3} = 44.72 \times \sqrt{460.84} = 960.01 m/s$$

Specific volume, $v_3 = v_1 \left(\frac{p_1}{p_3}\right)^{\frac{1}{n}} = 0.258 \left(\frac{10}{1}\right)^{\frac{1}{1.3}} = 1.5165 \text{ m}^3/\text{kg}$

$\therefore$ Area at exit, $A_3 = \frac{mv_3}{C_3} = \frac{1 \times 1.5165}{960.01} = 1.57967 \times 10^{-3} m^2 = \mathbf{15.7967 \text{ cm}^2}$ *Ans*

From the Mollier diagram,

$$h_{3isen} = 2585 \text{ kJ/kg}$$

$\therefore$ Isentropic heat drop, $(h_1 - h_{3isen}) = 3050 - 2585 = \mathbf{465 \text{ kJ/kg}}$ *Ans*

$\therefore$ Heat loss in availability, $dq = (h_1 - h_{3isen}) - (h_1 - h_3) = 465 - 460.84 = 4.16 \text{ kJ/kg}$

$\therefore$ Increase in enthalpy $= \int \frac{dq}{T} = \frac{4.16}{336.82} = \mathbf{0.01235 \text{ kJ/kg K.}}$ *Ans*

Problem 5.15 *Dry saturated steam with an initial pressure of 11 bar is expanded to 1.5 bar through a convergent divergent nozzle. The flow rate of steam is estimated as 2 kg/sec. area of cross section of nozzle as its throat is 1 cm^2. If the efficiency of nozzle is 88%. Find the numbers of nozzles and exact throat and exit area. Take the index of expansion as 1.2.* ***[May '13, 7 Marks]***

Solution : Given data (For reference refer Fig. 5.16)

$p_1 = 11 bar, p_3 = 1.5 bar, m = 2 kg/s, A_2 = 1 cm^2 = 1 \times 10^{-4} m^2$, $\eta_n = 88\%$, Number of nozzles = ?, A_2 = ?, A_3 = ?

We know that, $\frac{p_2}{p_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$

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Since the steam is dry and saturated, $n = 1.135$

$$\therefore \frac{p_2}{p_1} = \left(\frac{2}{1.135+1}\right)^{\frac{1.135}{1.135-1}} = 0.5774$$

$$\therefore p_2 = 0.5774\ p_1 = 0.5774 \times 11 = 6.3514\ bar$$

From the Mollier's diagram,

$$h_1 = 2780\ kJ/kg, h_2 = 2680\ kJ/kg, h_3 = 2440\ kJ/kg$$

From steam tables

$$v_{g2} = 0.297\ m^3/kg, \text{ (at 6.3514 bar)}, v_{g3} = 1.159\ m^3/kg \text{ (at 1.5 bar)}$$

Enthalpy at point 2′

$$h_2' = h_1 - \eta_n (h_1 - h_2) = 2780 - 0.88(2780 - 2680)$$
$$= 2692\ kJ/kg$$

Enthalpy at point 3′

$$h_3' = h_2' - \eta_n (h_2' - h_3) = 2692 - 0.88(2692 - 2440)$$
$$= 2470.24\ kJ/kg$$

From Mollier's chart

$$x_2' = 0.97,\ x_3' = 0.895$$

Specific volume of steam;

$$v_2' = x_2' v_{g2} = 0.97 \times 0.297 = 0.28809\ m^3/kg$$

$$v_3' = x_3' v_{g3} = 0.895 \times 1.159 = 1.0373\ m^3/kg$$

Actual velocity of steam at throat ;

$$C_2' = 44.72\sqrt{h_1 - h_2'} = 44.72\sqrt{2780 - 2692}$$
$$= 419.51\ m/s$$

Actual velocity of steam at exit

$$C_3' = 44.72\sqrt{h_1 - h_3'} = 44.72\sqrt{2780 - 2470.24}\ ;$$
$$= 787.072\ m/s$$

Mass of steam used per second per nozzle;

$$m_1 = \frac{A_2 C_2'}{v_2'} = \frac{1 \times 10^{-4} \times 419.51}{0.28809} = 0.1456\ kg/s$$

$$\therefore \text{Number of nozzles} = \frac{\text{Total mass flow rate}}{\text{Mass flow rate per nozzle}}$$

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$$= \frac{2}{0.1456} = 13.736 \text{ say } \mathbf{14}$$ **Ans**

Now, total mass flow rate = No. of nozzle x Exact mass flow rate per nozzle

$$2 = 14 \text{ x Exact mass flow rate per nozzle}$$

$\therefore$ Exact mass flow rate per nozzle, $m_1' = 0.14285 \ kg/s$

$$\therefore \text{Exact throat area, } A_2' = \frac{m_1' v_2'}{C_2'} = \frac{0.14285 \times 0.28809}{419.51'}$$

$$= 9.81 \times 10^{-5} m^2 = \mathbf{0.981 \ cm^2}$$ **Ans**

$$\text{Exact exit area, } A_3' = \frac{m_1' v_3'}{C_3'} = \frac{0.14285 \times 1.0373}{787.072}$$

$$= 1.8826 \times 10^{-4} m^2 = \mathbf{1.8826 \ cm^2}$$ **Ans**

Problem 5.16 *Dry saturated steam at pressure of 6 bar flow through nozzles at the rate of 4.5 kg/s and discharges at a pressure of 1.6 bar. The loss due to friction occurs only in the diverging portion of the nozzle and its magnitude is 12% of the total isentropic enthalpy drop. Assume the isentropic index of expansion n = 1.135, determine the cross sectional area at the throat and exit of the nozzles.* ***[Nov. '14, 7 Marks]***

Solution : Given data (refer Fig. 5.20)

p_1 = 6 bar, m = 4.5 kg/s, p_3 = 1.6 bar, loss = 12% of isentropic enthalpy drop, n = 1.135, A_2 = ?, A_3 = ?

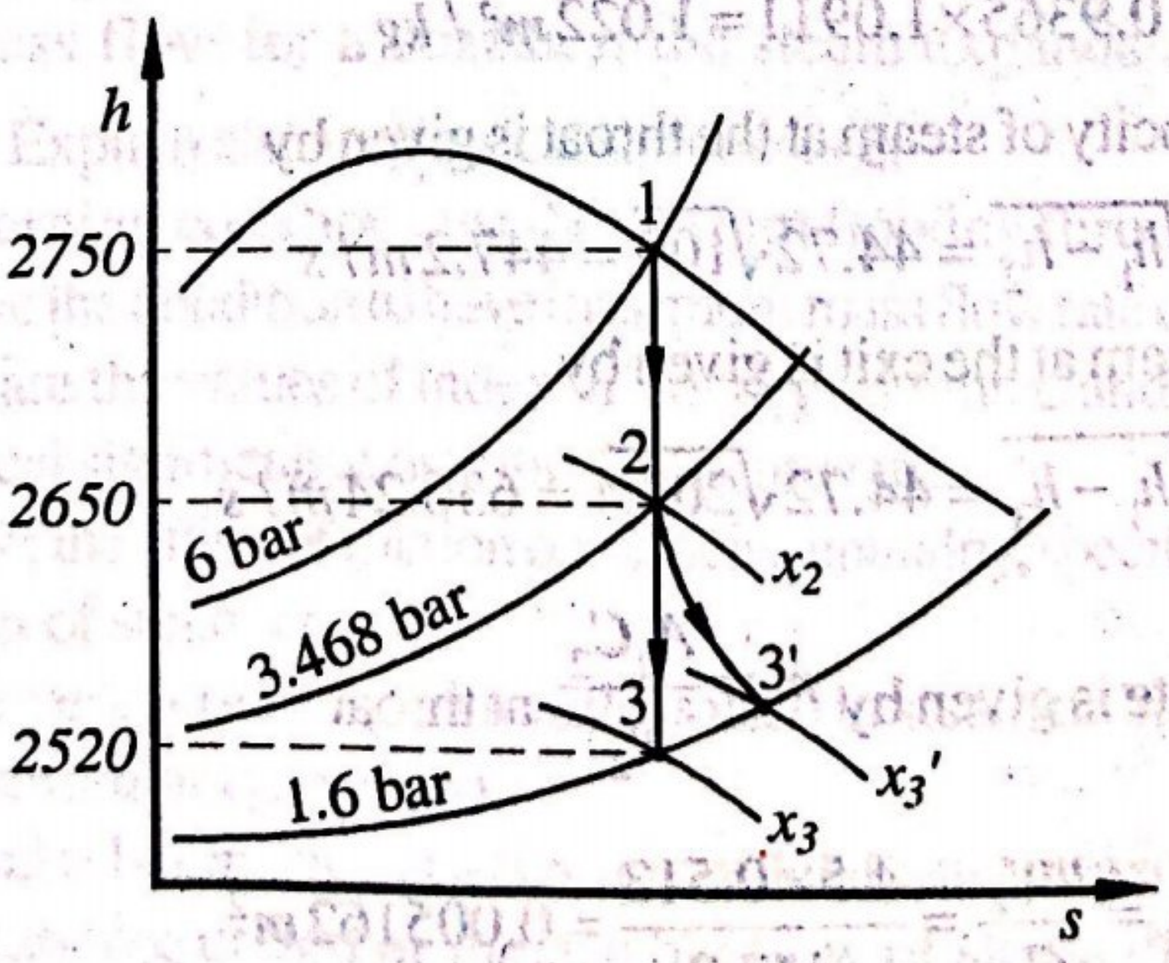


Fig. 5.20

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we know that $\frac{p_2}{p_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} = \left(\frac{2}{1.135+1}\right)^{\frac{1.135}{1.135-1}}$

$= 0.578$

Therefore, $p_2 = 0.578$, $p_1 = 0.578 \times 6 = 3.468$ bar.

Reading from the Mollier diagram, the isentropic enthalpy change from 6 bar to 1.6 bar.

$$(h_1 - h_3)_{isen.} = 230\ kJ/kg$$

Therefore, $(h_1 - h_3') = 0.88 \times 230 = 202.4\ kJ/kg$

Also from the Mollier diagram, the dryness fraction at point 2 and 3′ is respectively $x_2 =$ 0.964 and $x_3' = 0.9365$.

Isentropic enthalpy change from 6 bar (entrance) to 3.468 bar (throat) as read from the Mollier diagram.

$$(h_1 - h_2) = 100\ kJ/kg$$

For accurate values of dry saturated specific volumes, use steam tables.

$$v_{g2} = 0.5312 m^3/kg,\ v_{g3}' = 1.0911 m^3/kg$$

$$v_2 = x_2 v_{g2} = 0.964 \times 0.5321 = 0.513 m^3/kg$$

$$v_3' = x_3' v_{g2} = 0.9365 \times 1.0911 = 1.022 m^3/kg$$

Also, the velocity of steam at the throat is given by

$$C_2 = 44.72\sqrt{h_1 - h_2} = 44.72\sqrt{100} = 447.2\ m/s$$

and the velocity of steam at the exit is given by

$$C_3' = 44.72\sqrt{h_1 - h_3'} = 44.72\sqrt{202.4} = 636.24\ m/s$$

Also the mass flow rate is given by $m = \frac{A_2 C_2}{v_2}$ at throat.

Therefore, $A_2 = \frac{m v_2}{C_2} = \frac{4.5 \times 0.513}{447.2} = 0.005162 m^2$

$= 51.62\ cm^2$

Ans

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$$m = \frac{A_3 C_3'}{v_3'} \text{ at exit.}$$

$$\therefore A_3 = \frac{m v_3'}{C_3'} = \frac{4.5 \times 1.022}{636.24} = 0.007228\, m^2$$

$$= 72.28 \text{ cm}^2 \quad \textit{Ans}$$

Exercise

5.1 What are the functions of steam nozzle ?

5.2 Explain the functions of the convergent portion, the throat and the divergent portion of a convergent-divergent nozzle with reference to flow of steam.

5.3 Describe different types of nozzle with neat sketch.

5.4 Derive an expression for velocity of steam at exit of nozzle.

5.5 Define the term 'nozzle efficiency'.

5.6 Derive an expression for mass flow rate of steam through the nozzle.

5.7 Starting from fundamentals, show that for maximum discharge through nozzle, the ratio of throat pressure to inlet pressure is given by $\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$ where 'n' is the index of isentropic expansion of steam through the nozzle.

5.8 What is critical pressure ? Derive the expression of critical pressure ratio to have maximum mass flow for a nozzle, if the steam expands according to $pv^n = C$ isentropically. Explain also its physical significance.

5.9 State the governing equations used for steam flowing through nozzle. Using these equation, derive the condition to have maximum mass flow rate of steam, flowing through nozzle. What are the values of index 'n' for dry-saturated and for superheated steam ?

5.10 Explain physical significance of critical pressure ratio.

5.11 Discuss briefly : the effect of friction on velocity, enthalpy, specific volume, and condition of steam at exit of steam nozzle.

5.12 Why is the convergent-divergent nozzle is used in steam turbine ? How will you select a nozzle for particular type of job ?

5.13 Explain general relationship between area-velocity and pressure in nozzle flow.

5.14 Explain the supersaturated or metastable flow of steam through a nozzle and the significance of Wilson's line.

5.15 What are the effects of supersaturation on discharge and heat drop ?

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5.16 Explain the process of super-saturation in the steam nozzle with the help of H-S diagram. Also define degree of undercooling and degree of supersaturation.

5.17 Explain the effect of variation of back pressure of nozzle.

Unsolved Numerical Problems

5.1 Steam at a pressure of 8 bar and 200°C is supplied to a convergent divergent nozzle with a throat area of 1400 mm^2. The exit is below critical pressure. Determine the coefficient of discharge, if the flow is 5500 kg of steam per hour. **[Ans :** 0.934]

5.2 Estimate the mass flow rate of steam in a nozzle with the following data :
Inlet conditions = 10 bar pressure and 80°C superheat
Back pressure = 0.5 bar
Throat diameter = 12 mm **[Ans :** 0.131 kg/s]

5.3 Dry saturated steam at a pressure of 10 bar enters a convergent-divergent nozzle and leaves it at a pressure of 1.5 bar. If the flow is isentropic, and the corresponding expansion index is 1.135; find the ratio of cross-sectional area at exit and throat for maximum discharge. **[Ans :** 2.055]

5.4 Steam enters a group of nozzles of a steam turbine at 15 bar and 250°C and leaves at 1.5 bar. The steam turbine develops 250 kW with a specific steam consumption of 12 kg/kWh. If the diameter of nozzles at throat is 8 mm, calculate the number of nozzles. **[Ans :** 10]

5.5 Steam expands in a set of nozzles from 12 bar, 300°C to 1 bar. The convergent part of the nozzles are sharp and frictionless. In the divergent part, the friction loss may be taken as 14% of the isentropic enthalpy drop. If the steam flow rate is 1 kg/s and the initial velocity of steam is negligible, find the minimum area of the nozzles. If the exit diameter of nozzles is 30 mm, find the number of nozzles.
[Ans : $A_2 = 6.2366\ cm^2$, 3 nozzles]

5.6 Steam at a pressure of 12 bar and 0.95 dry discharges through a nozzle having throat area of 430 mm^2. If the back pressure is 1 bar, find (i) final velocity of the steam, and (ii) cross sectional area of the nozzle at exit for maximum discharge.
[Ans : 883.15 m/s, (ii) 1093.61 mm^2]

5.7 A nozzle is supplied with steam at 7 bar and temperature 300°C and discharge is at 1 bar. The diverging portion is 50 mm long and throat diameter is 5 mm. Assuming 10% of the total enthalpy drop to be lost in friction in the diverging portion of nozzle, find the cone angle of the divergent portion. **[Ans :** $2\theta = 2.2°$]

5.8 A nozzle is supplied with steam at a rate of 1 kg/s from inlet condition of 10 bar dry saturated and exit pressure of 1 bar. The isentropic efficiencies for the convergent section and divergent section are 0.95 and 0.9 respectively. Determine : (i) the throat and exit diameters, (ii) length of divergent section if the semicone angle of the divergent section (θ) is 7°, (iii) power in kW corresponding to the exit velocity of steam

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[**Ans :** (i) $D_2 = 2.27$ cm, $D_3 = 3.2$ cm, (ii) L = 3.8 cm, (iii) 9117 kW]

5.9 An impulse turbine having a set of 24 nozzles receives steam at 20 bar, 400°C. The pressure of steam at exit is 11 bar. If the total discharge is 300 kg/min and nozzle efficiency is 92%, determine the cross-sectional area of the exit of each nozzle. If the steam has a velocity of 70 m/s at entry to the nozzles, find the percentage increase in discharge. [**Ans :** $8.4\times10^{-5}\, m^2$, 0.72%]

5.10 A convergent-divergent nozzle is supplied with steam at a pressure of 10 bar and 230°C. Supersaturated expansion occurs according to $pv^{1.3} = c$ in the nozzle down to an exit pressure of 3.2 bar. The exit diameter of the nozzle is 25 mm. If the flow through the nozzle is chocked, find (i) exit velocity, (ii) mass flow rate, and (iii) throat diameter. [**Ans :** (i) 662.95 m/s, (ii) 0.6178 kg/s, (iii) 0.0235 m]

5.11 A convergent-divergent nozzle is supplied with steam at a pressure of 12 bar and 250°C. The exit pressure is 2 bar. The condensation does not take place while the steam is in the nozzle due to supersaturation. Determine the throat area, degree of undercooling, degree of supersaturation exit area and actual enthalpy drop for mass flow rate of 1 kg/s. [**Ans** : 5.998 cm^2, 47.33°C, 5.714, 9.27 cm^2, 339.16 kJ/kg]

Objective Type Questions

1. The main function of steam nozzle is to convert
   (a) pressure and heat energy of steam into mechanical work.
   (b) kinetic energy of steam into mechanical work.
   (c) pressure and heat energy of steam into kinetic energy of steam.
   (d) kinetic energy of steam into pressure energy of steam.
2. It is required to increase the velocity of steam from negligible to supersonic velocity, which nozzle you will select.
   (a) convergent (b) convergent-divergent
   (c) divergent (d) all of the above
3. The velocity of steam leaving the nozzle is given by
   (a) $44.72\sqrt{h_1 - h_2}$ (b) $44.72\sqrt{h_2 - h_1}$
   (c) $72.44\sqrt{h_1 - h_2}$ (d) $72.44\sqrt{h_2 - h_1}$
4. What is (are) the effect(s) of friction on nozzle performance ?
   (a) reduction in final velocity (b) increase in final dryness fraction
   (c) increase in critical pressure (d) all of the above
5. The mass flow rate through nozzle is given by

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(a) $m = A_2\sqrt{\frac{2n}{n-1}\frac{p_2}{v_2}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right]}$

(b) $m = \sqrt{\frac{2n}{n-1}\frac{p_1}{v_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right]}$

(c) $m = \sqrt{\frac{n}{n-1}\frac{p_1}{v_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right]}$

(d) $m = \sqrt{\frac{n}{n-1}\frac{p_2}{v_2}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right]}$

6. The critical pressure ratio for maximum mass flow rate per unit area of nozzle is given by

(a) $\left(\frac{2}{n+1}\right)^{\frac{n}{n+1}}$ (b) $\left(\frac{2}{n-1}\right)^{\frac{n}{n+1}}$ (c) $\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$ (d) $\left(\frac{2}{n-1}\right)^{\frac{n}{n-1}}$

7. The critical pressure ratio for initially dry saturated steam is
(a) 0.5774 (b) 0.5457 (c) 0.548 (d) 0.59

8. The critical pressure ratio for initially superheated steam is
(a) 0.5774 (b) 0.5457 (c) 0.548 (d) 0.59

9. The flow of steam is supersonic
(a) in the convergent portion of nozzle. (b) at the throat of the nozzle.
(c) at the inlet to the nozzle. (d) in the divergent portion of the nozzle.

10. The difference of supersaturated temperature and saturation temperature at that pressure is known as
(a) degree of superheat (b) degree of supersaturation
(c) degree of undercooling (d) all of the above

11. In a nozzle, the effect of supersaturation is to
(a) increase the dryness fraction of steam
(b) reduce enthalpy drop.
(c) increase the entropy and specific volume of steam
(d) all of the above.

12. The critical pressure ratio of convergent nozzle is defined as
(a) The ratio of inlet pressure to outlet pressure only when mass flow rate per unit area is maximum.

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(b) The ratio of outlet pressure to inlet pressure only when mass flow rate per unit area is maximum.
(c) The ratio of outlet pressure to inlet pressure.
(d) The ratio of outlet pressure to inlet pressure only when mass flow rate per unit area is minimum.

13. When the back pressure of a nozzle is below the designed value of pressure at exit of the nozzle, the nozzle is said to be
(a) over expanded (b) under expanded
(c) chocked (d) none of the above

14. Due to supersaturation in steam nozzle
(a) actual mass flow rate of steam is less than calculated value.
(b) actual mass flow rate of steam is equal to calculated value.
(c) actual mass flow rate of steam is more than calculated value.
(d) none of the above.

Answers

1. (c)	**2.** (b)	**3.** (a)	**4.** (d)	**5.** (b)	**6.** (c)	**7.** (a)
8. (b)	**9.** (d)	**10.** (c)	**11.** (d)	**12.** (b)	**13.** (b)	**14.** (c)

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GTU Study Materials ☑

Chapter 6
Steam Turbine

6.1 Introduction
6.2 Principle of operation
6.3 Classification of steam turbines
6.4 Compounding of steam turbines
6.4.1 Pressure compounded impulse turbine (Rateau Turbine)
6.4.2 Velocity compounded impulse turbine (Curtis turbine)
6.4.3 Pressure -Velocity compounded of impulse turbine
6.5 Impulse turbine - velocity diagram
6.5.1 Calculations for work power and efficiencies
6.6 Condition for maximum blade efficiency of impulse turbines
6.7 Reaction Turbine
6.8 Velocity diagram for reaction turbine
6.9 Degree of reaction
6.10 Parson's turbine
6.11 Condition for maximum efficiency of Parson's reaction turbine
6.12 Reheat-factor
6.13 Internal and other efficiencies
6.14 Governing of steam turbines
6.14.1 Throttle governing
6.14.2 Nozzle governing
6.14.3 By-pass governing
6.14.4 Comparison of throttle and nozzle governing
6.15 Methods of attachment of blades to turbine rotor
6.16 Labyrinth packing
6.17 Losses in steam turbines
6.17.1 Internal losses
6.17.2 External losses
6.18 Special types of steam turbines
6.19 Back pressure turbines
6.20 Pass out turbine or Extraction turbine
6.21 Mixed pressure turbine

6.1 Introduction

*A **steam turbine** is a prime mover which continuously converts the energy of high pressure, high temperature steam supplied by a steam generator into shaft work with the low temperature steam exhausted to a condenser.* This energy conversion essentially occurs in two steps :

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(1) The high pressure, high temperature steam first expands in efficient nozzles and comes out at a high velocity.

(2) The high velocity jets of steam coming out of the nozzles, impinge on the blades mounted on a wheel, get deflected by an angle and suffer a loss of momentum which is absorbed by the rotating wheel in producing torque.

A steam turbine is basically an assemblage of nozzles and blades.

The steam turbines offer following advantages over reciprocating steam engines :

(1) A steam turbine has higher thermal efficiency than reciprocating steam engines.

(2) The power delivered by the turbine is uniform, hence it does not need any flywheel.

(3) The steam turbines can develope high rotational speed (18000-24000 rpm) but such high speed is not possible in the case of reciprocating steam engines.

(4) Steam turbines are perfectly balanced and hence present minimum vibrational problem.

(5) It needs no internal lubrication due to the absence of rubbing parts.

(6) It is compact and it has low weight to power ratio.

6.2 Principle of operation [May '13]

If high-velocity steam from nozzle exit is allowed to blow on to a curved blade, the steam will suffer a change in direction as it passes across the blade and leaves it as shown in Fig. 6.1.

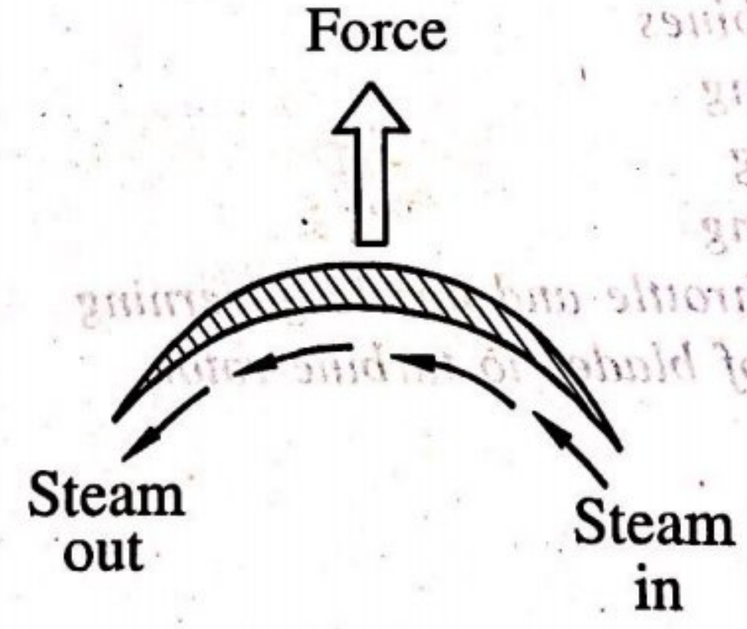


Fig. 6.1 Principle of operation of steam turbine

As a result of its change in direction across the blade, the steam will impart a force to the blade. This force will act in the direction shown in Fig. 6.1.

Now if the blade were free to move, it would move in the direction of force as depicted. If, therefore, a number of blades were fixed on the circumference of a disc which is free to rotate on a shaft, then steam blown across the blades in the way described would cause the disc to rotate. This is the working principle of a steam turbine.

The arrangement of simple impulse turbine is shown in Fig. 6.2. It mainly consists of a nozzle or a set of nozzles, a rotor mounted on a shaft, one set of moving blades attached to the rotor and a casing. The uppermost portion of the diagram shows a longitudinal section through the upper half of the turbine, the middle portion shows the development of the nozzles and

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blading i.e. the actual shape of the nozzle and blading, and the bottom portion shows the variation of absolute velocity and absolute pressure during flow of steam through passage of nozzles and blades. The principle example of this type of turbine is the ***De-Laval turbine***.

It is obvious from the Fig. 6.2 that the complete expansion of steam from the boiler pressure to the exhaust pressure or condenser pressure takes place only in one set of nozzles i.e. the pressure drop takes place only in nozzles. It is assumed that the pressure in the recess between nozzles and blades remains the same. The steam at condenser pressure or exhaust pressure enters the blade and comes out at the same pressure i.e. the pressure of steam in the blade passages remains approximately constant and equal to the condenser pressure.

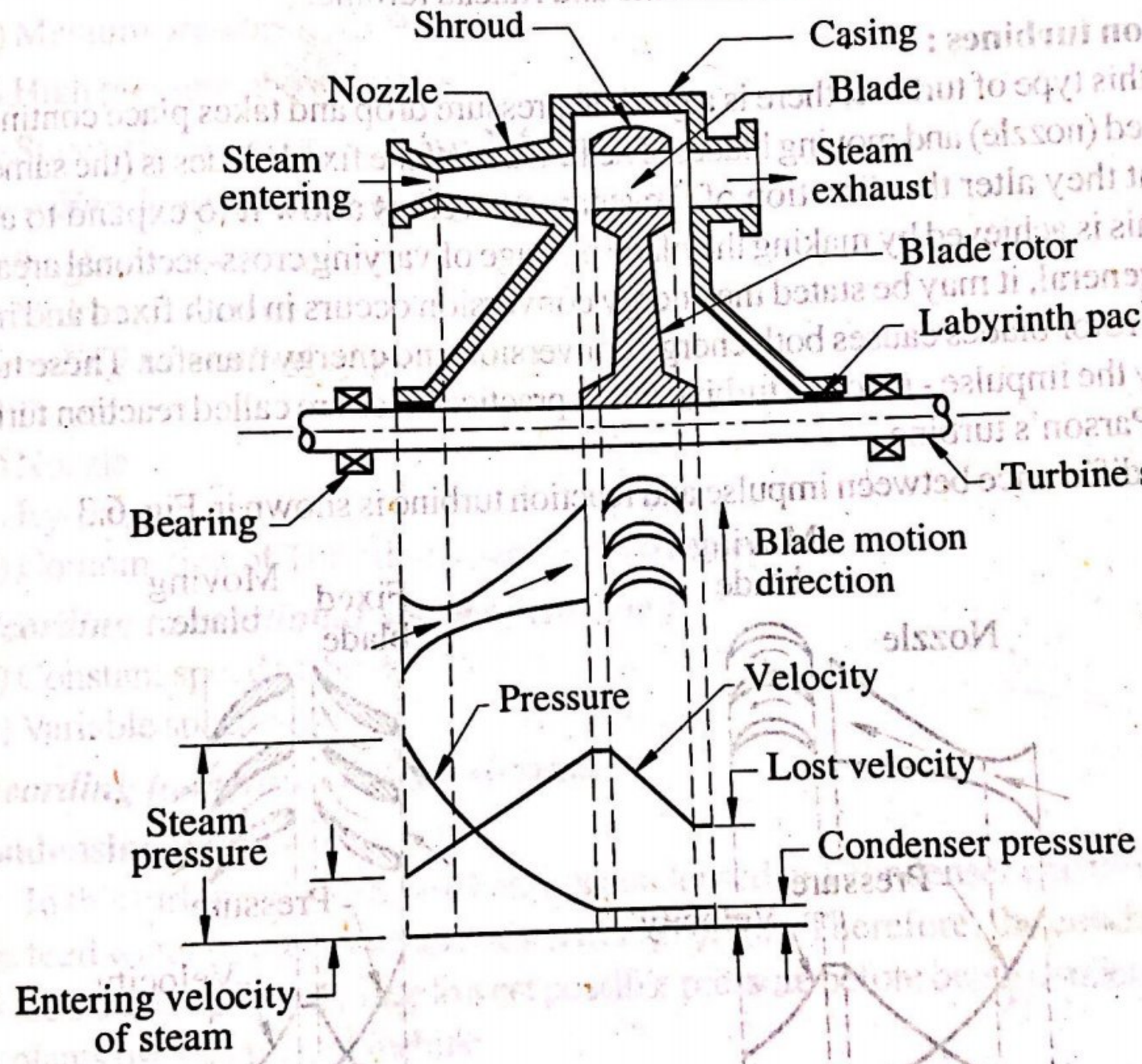


Fig. 6.2 Arrangement of simple impulse turbine

The steam leaves the nozzles at a very high velocity (supersonic) of about 1100 m/s. The steam at such high velocity enters the blades and reduces along the passage of blades and comes out with an appreciable amount of velocity. The loss of energy due to this high exit velocity is commonly the ***carry over loss*** or ***leaving loss***.

6.3 Classification of steam turbines [May '14]

The steam turbines can be classified as follows :

(1) According to principle of operation :

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(a) Impulse turbine :

If the flow of steam through the nozzles and moving blades of a turbine takes place in such a manner that the drop in steam pressure takes place only in nozzles and not in moving blades, such a turbine is called impulse turbine. In other words, in impulse turbine, the steam is expanded only in nozzles and pressure at the outlet side of the blades is equal to that at inlet side. This is obtained by making the blade passage of constant cross-section area.

As a general statement it may be stated that energy conversion takes place only in nozzles, and moving blades only cause energy transfer.

Examples : De-Laval turbine, Curtis turbine and Rateau turbine.

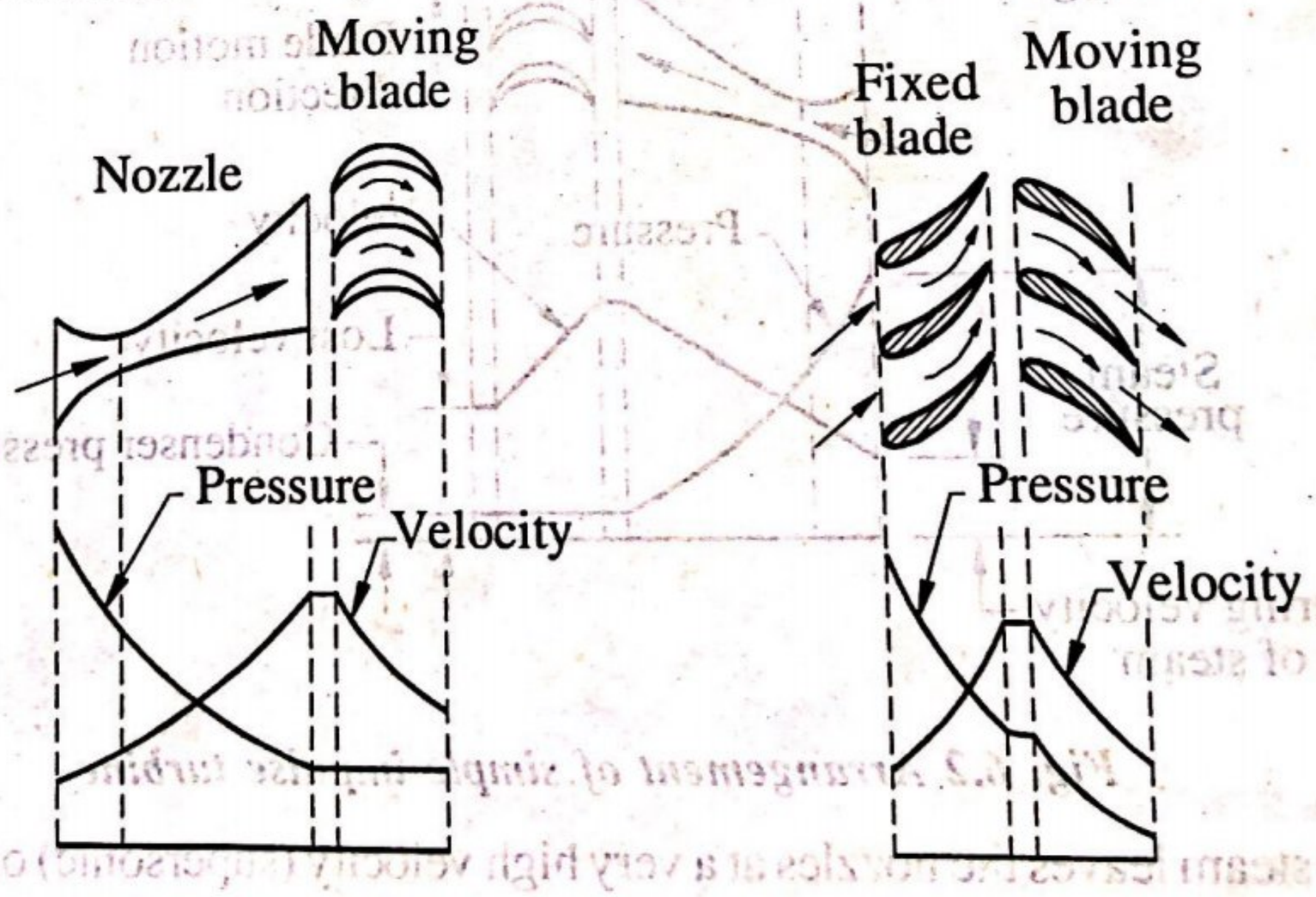
(b) Reaction turbines :

In this type of turbine, there is a gradual pressure drop and takes place continuously over the fixed (nozzle) and moving blades. The function of the fixed blades is (the same as the nozzle) that they alter the direction of the steam as well as allow it to expand to a larger velocity. This is achieved by making the blade passage of varying cross-sectional area.

In general, it may be stated the energy conversion occurs in both fixed and moving blades. The rotor blades causes both energy conversion and energy transfer. These turbines are basically the impulse - reaction turbine but in practice these are called reaction turbines.

Example : Parson's turbine.

The difference between impulse and reaction turbine is shown in Fig. 6.3



Impulse turbine ***Reaction turbine***

Fig. 6.3 Difference between impulse turbine and reaction turbine

(2) According to direction of steam flow :

(a) Axial flow turbine :

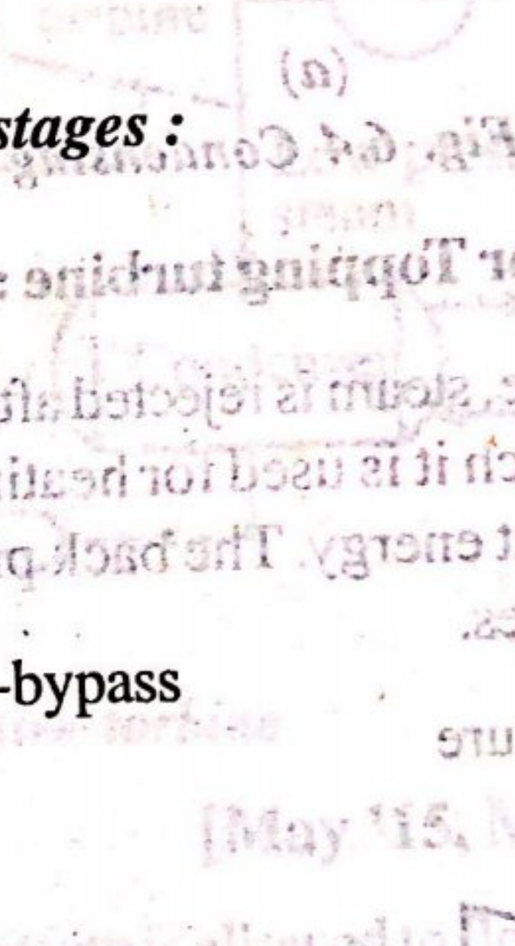
In axial flow turbine, the steam flows along the axis of the shaft. It is the most suitable

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turbine for large turbo generators and that is why it is used in all modern steam power plants.

(b) Radial flow turbine :

In this turbine, the steam flows in radial direction. It incorporates two shafts end to end, each driving a separate generator.



(c) Tangential flow turbine :

In this type, the steam flows in the tangential direction. This turbine is very robust but not particularly efficient machine.

(3) According to the inlet steam pressure :

(a) Low pressure upto 2 bar

(b) Medium pressure upto 50 bar

(c) High pressure above 50 bar

(d) Supercritical pressure above 225 bar

(4) According to number of energy conversion stages :

(a) Single stage turbine

(b) Multistage turbine

(5) According to method of governing :

(a) Throttle

(b) Nozzle

(c) By-pass

(d) Combination of Throttle-bypass or Nozzle-bypass

(6) According to rotational speed of turbine :

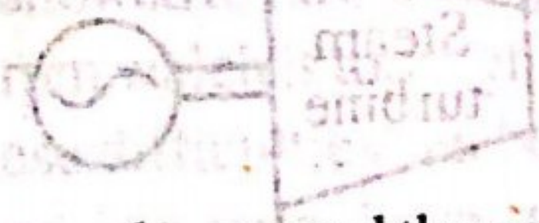
(a) Constant speed turbines

(b) Variable speed turbines

(7) According to means of heat rejection :

(a) Condensing turbine :

In this turbine, the exhaust steam is condensed in a condenser and the condensate is used as feed water in the boiler as shown in Fig. 6.4(a). Therefore, the condensing turbine allows the steam to expand to the lowest possible pressure before being condensed. All steam power plants use this type of turbine.



(b) Non condensing turbine :

When the exhaust steam coming out from the turbine is not condensed but exhausted in the atmosphere is called non-condensing turbine (Fig. 6.4(b)). The exhaust steam is not recovered for feed water in the boiler.

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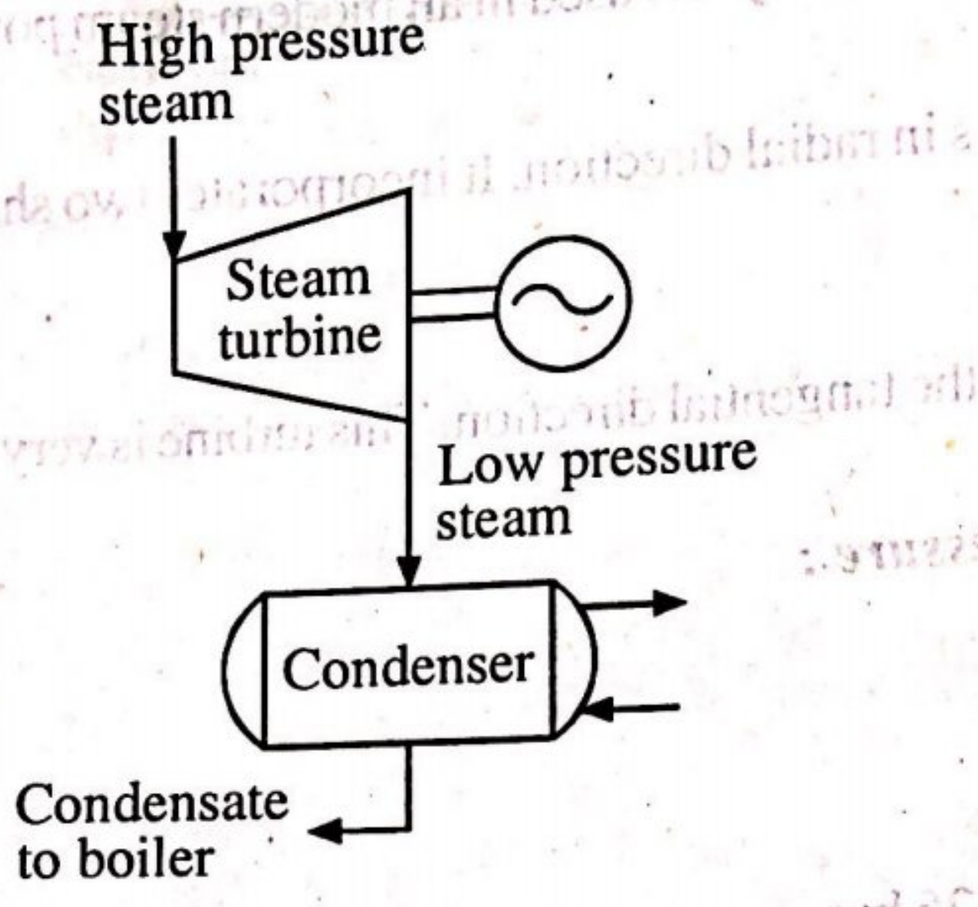


Fig. 6.4 Condensing and non-condensing turbine

(c) Back pressure or Topping turbine :

In this turbine, steam is rejected after expansion to the lowest suitable pressure (above atmospheric) at which it is used for heating purpose. Thus back pressure turbine supplies power as well as heat energy. The back pressure turbine as shown in Fig. 6.5(a) generally used in sugar industries.

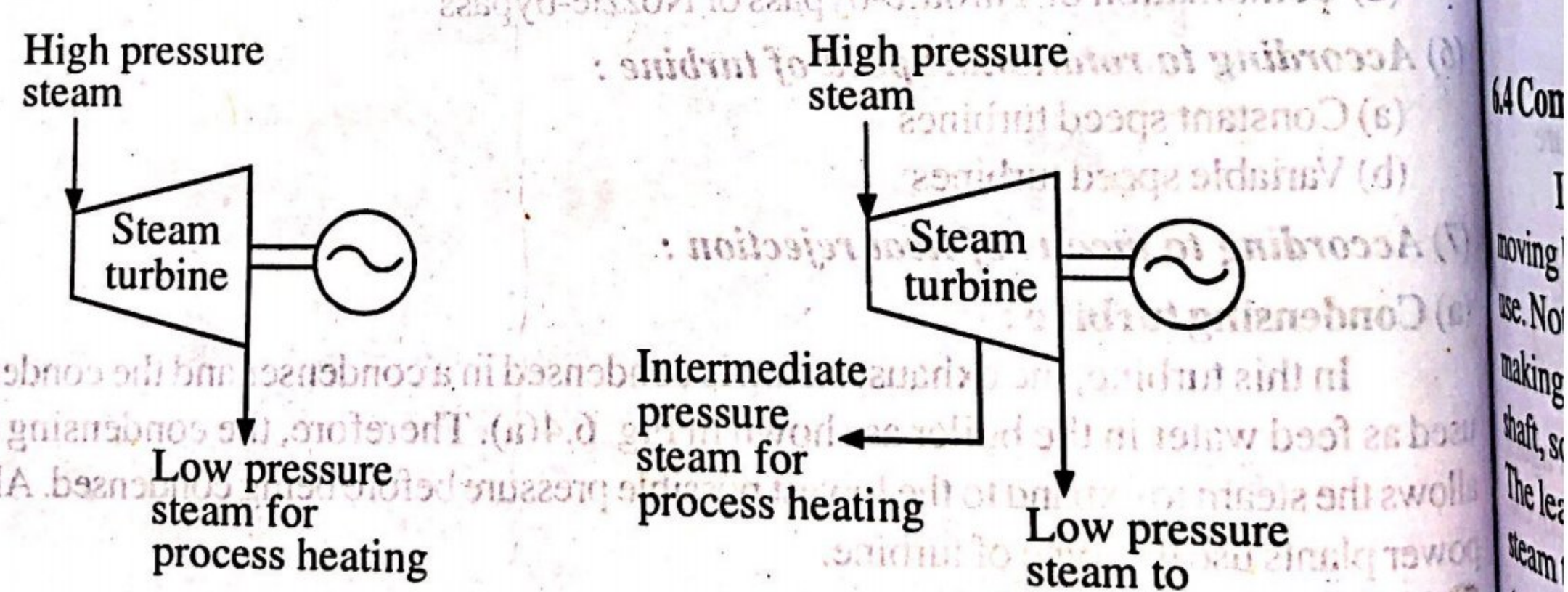


Fig. 6.5 Back pressure and pass-out turbine

(d) Pass-out turbine :

In this turbine, a considerable proportion of the steam is extracted from some suitable point in the turbine where the pressure is sufficient for use in process heating; the remainder continuing through the turbine. The arrangement of this turbine is shown in Fig. 6.5(b). This

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type of turbine is suitable where there is dual demand of steam – one for power and the other for industrial process heating e.g. sugar industries.

(e) Regenerative turbine :

The arrangement of regenerative turbine is shown in Fig. 6.6. This turbine incorporates a number of extraction branches, through which small proportions of the steam are continuously extracted for the purpose of heating the boiler feed water in a feed water heater in order to increase the thermal efficiency of the plant.

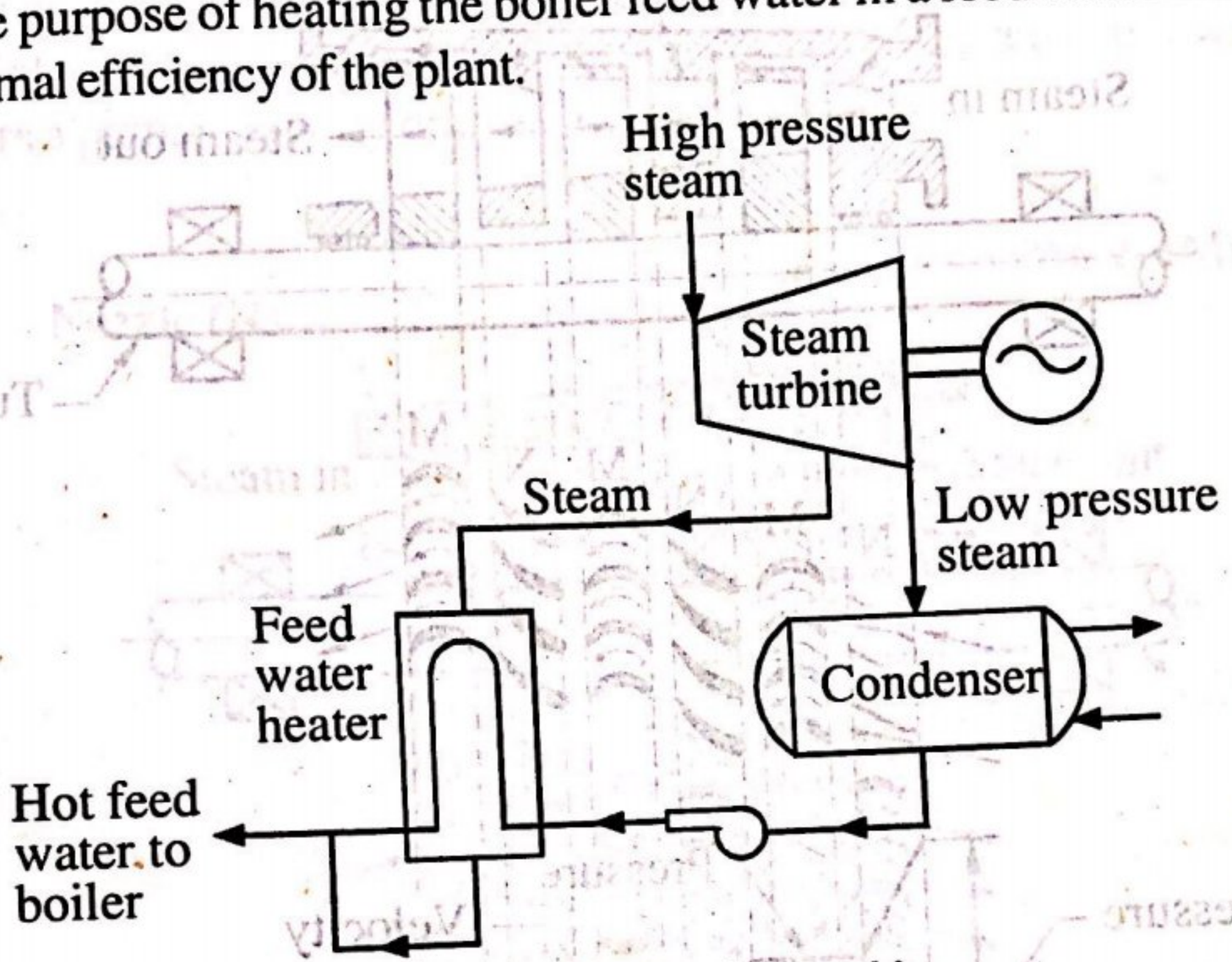


Fig. 6.6 Regenerative turbine

6.4 Compounding of steam turbines [May '15, May '13, May '12]

In a simple impulse turbine, high velocity steam is allowed to flow through one row of moving blades, it produces a rotor speed of about 30,000 rpm which is too high for practical use. Not only this, the leaving loss is also very high. These difficulties can be reduced by making use of more than one set of nozzles, blades, rotors, in a series keyed to a common shaft, so that either the steam pressure or the jet velocity is absorbed by the turbine in stages. The leaving loss also will then be less. This process is called ***compounding*** or ***staging*** of steam turbines. *Compounding is a method for reducing the rotational speed of the impulse turbine to practical limits.*

There are three main types of compounding :

(1) Pressure compounded impulse turbine,

(2) Velocity compounded impulse turbine,

(3) Pressure and velocity compounded impulse turbine.

6.4.1 Pressure compounded impulse turbine (Rateau Turbine) [Nov. '14]

In this type of turbine, the total pressure drop (steam generator pressure to condenser pressure) is divided into number of stages. Each turbine stage is provided with one raw of

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fixed blades which work as nozzles followed by a row of moving blades as shown in Fig. 6.7.

Steam enters the first row of nozzles where it suffers a partial drop of pressure and in result of that its velocity increases. The high velocity steam passes on to the first raw of moving blades where its velocity is reduced as shown in Fig. 6.7.

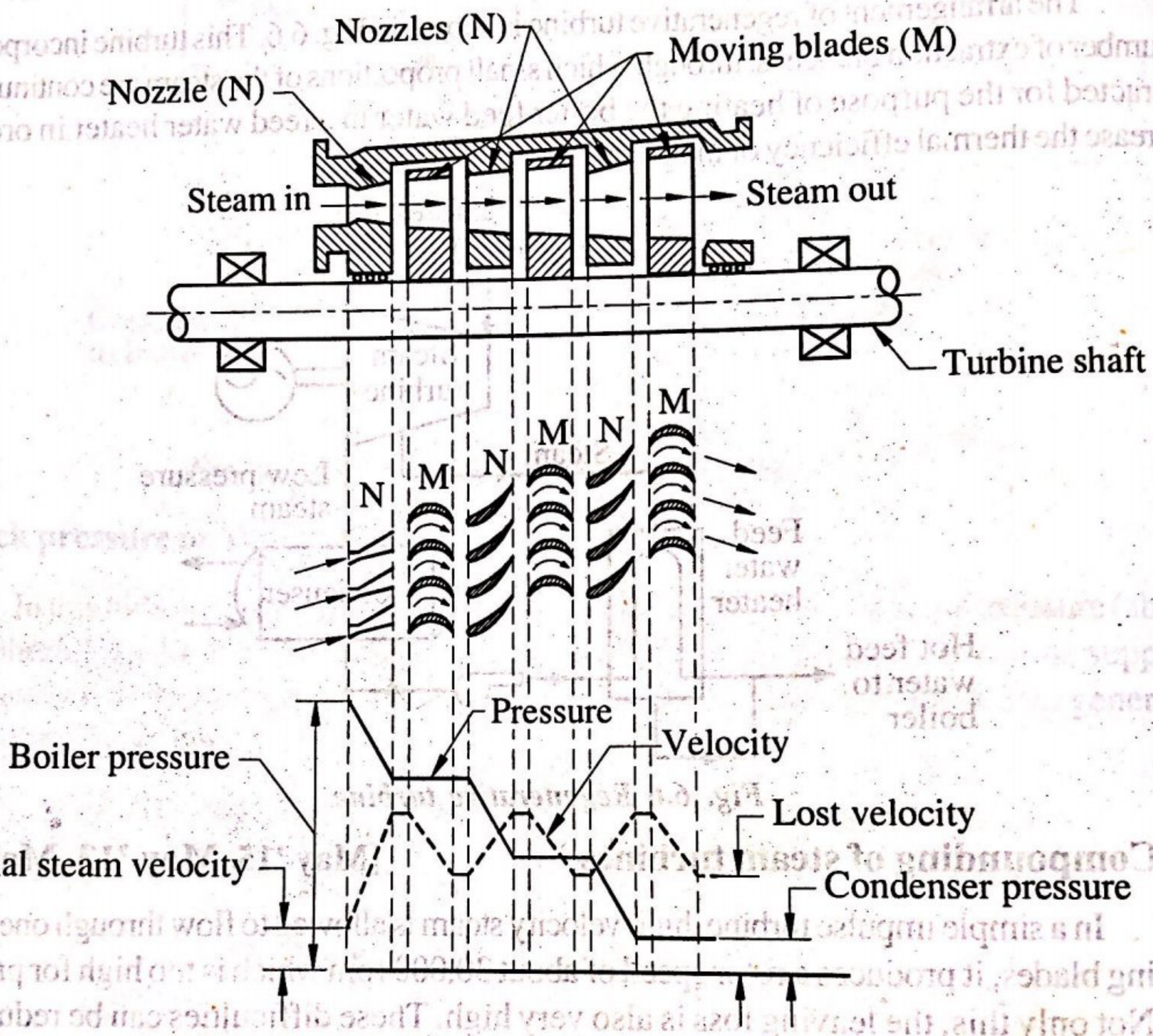


Fig. 6.7 Pressure compounding of impulse turbine

The steam then passes into the second row of nozzles where its pressure is again partially reduced and velocity is again increased. This high velocity steam passes from the nozzles to the second row of moving blades where its velocity is again reduced.

Thus pressure drop takes place in successive stages. Since a partial drop of pressure takes place in each stage, the steam velocities will not be so high with the effect that the turbine will run slower.

Advantages :

(1) The pressure compounding is quite efficient.

(2) It has constant blade velocity to steam velocity ratio.

Disadvantages :

(1) It needs more space since high number of stages are required.

(2) Its initial cost is high

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6.4.2 Velocity compounded impulse turbine (Curtis turbine) [May '14, May '12]

In velocity compounded impulse turbine, the pressure drop is full in one row of nozzles like that in simple impulse turbine but the resulting kinetic energy is absorbed by the rotor in a number of rows of moving blades with a ring of fixed blades in between two rows of moving blades. The fixed blades are mounted on the casing, while the moving blades are keyed in series on a common shaft. The function of the fixed blades is to correct the direction of entry of steam to the next row of moving blades.

The high velocity steam leaving the nozzles passes on to the first row of moving blades where it suffers a partial velocity drop as shown in Fig. 6.8.

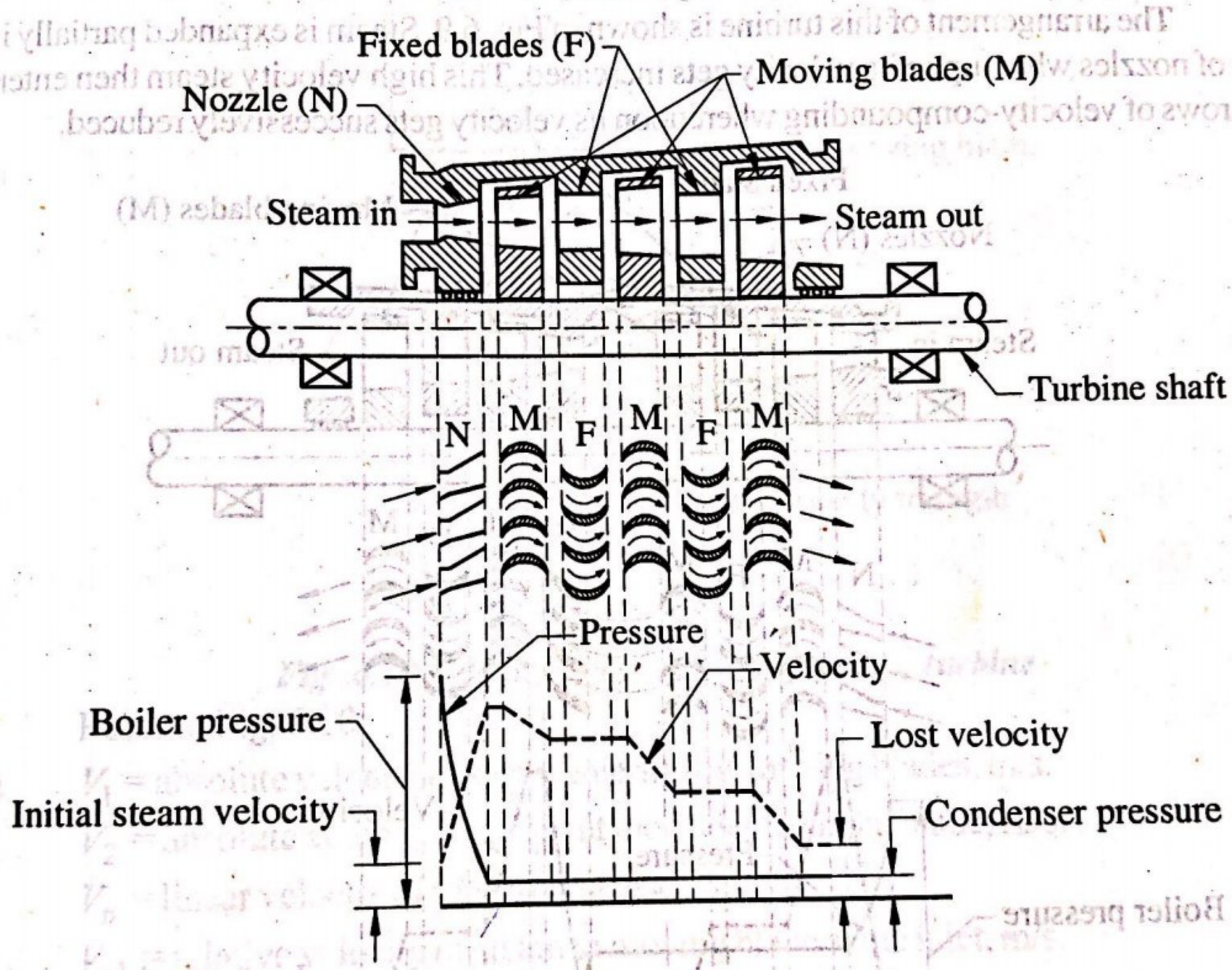


Fig. 6.8 Velocity compounding of impulse turbine

The direction of steam is then corrected by the next row of fixed blades and then it enters the second row of moving blades. Here the steam velocity is again partially reduced. Since only part of the velocity of the steam is used up in each row of the moving blades, a slower turbine results.

Advantages :

(1) It requires less number of stages and less space.

(2) The pressure drop takes place in nozzles, hence turbine casing has to withstand low pressure.

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(3) It's initial cost is low.

(4) Its starting is easier.

Disadvantages :

(1) Due to high velocity of steam friction losses are more.

(2) Its efficiency is low and keeps on decreasing with number of stages.

(3) Blade velocity to steam velocity ratio is not optimum for all the rows.

6.4.3 Pressure - Velocity compounded of impulse turbine

To take advantage of both pressure compounding and velocity compounding, a combination of pressure and velocity compounding is used.

The arrangement of this turbine is shown in Fig. 6.9. Steam is expanded partially in a row of nozzles whereupon its velocity gets increased. This high velocity steam then enters a few rows of velocity-compounding whereupon its velocity gets successively reduced.

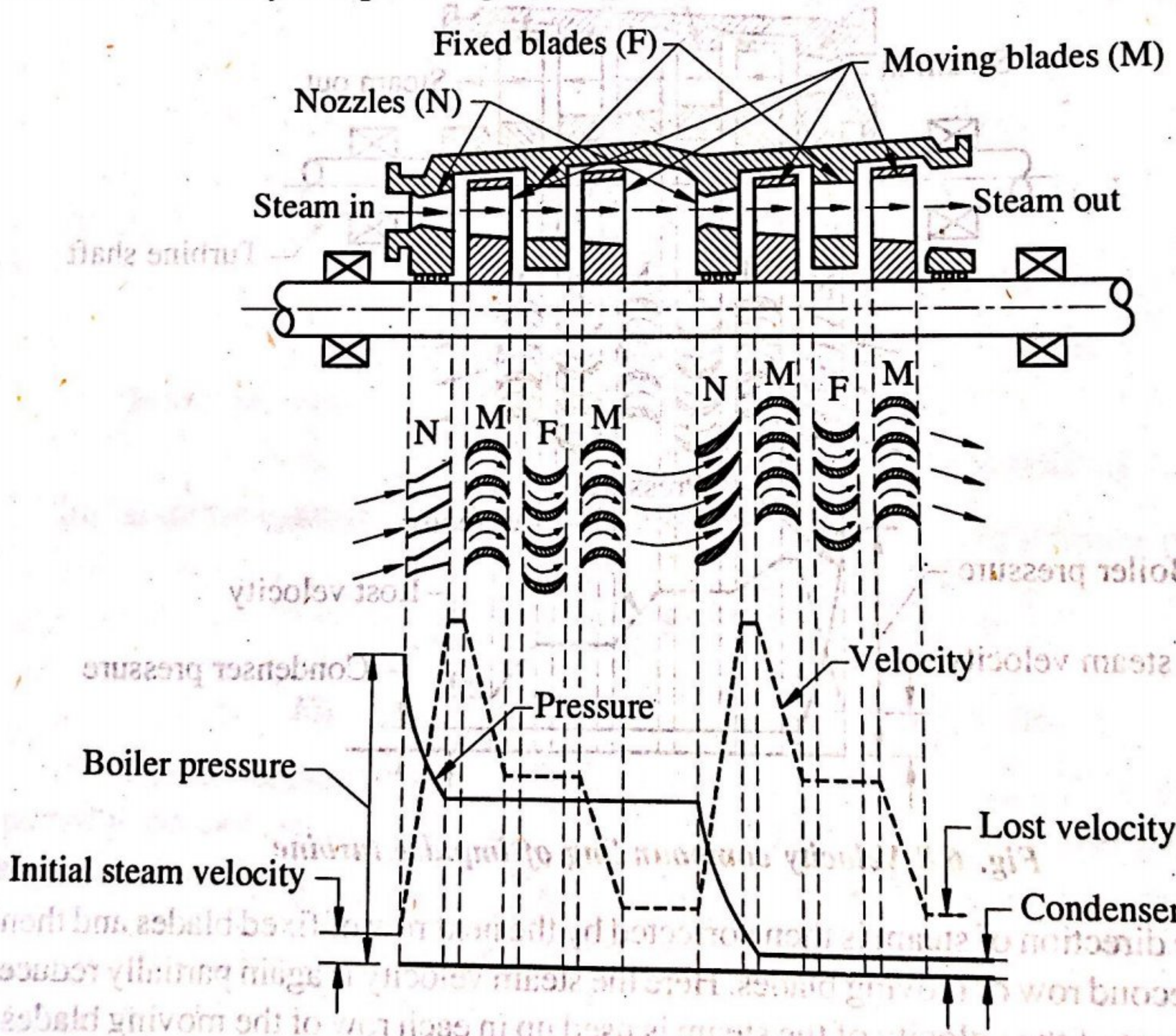


Fig. 6.9 Pressure-velocity compounding of impulse turbine

The velocity of the steam is again increased in the subsequent row of nozzles and then it is allowed to pass onto another set of velocity compounding that brings about a stagewise reduction of velocity of the steam.

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6.5 Impulse turbine - velocity diagram [Sep. '13, May '13]

We have already discussed that in an impulse turbine, the steam jet after leaving the nozzle impinges on one end of the blade. The jet then glides over the inside surface of the blade and finally leaves from the other edge as shown in Fig. 6.10. It may be noted that the jet enters and leaves the blades tangentially for shockless entry and exit.

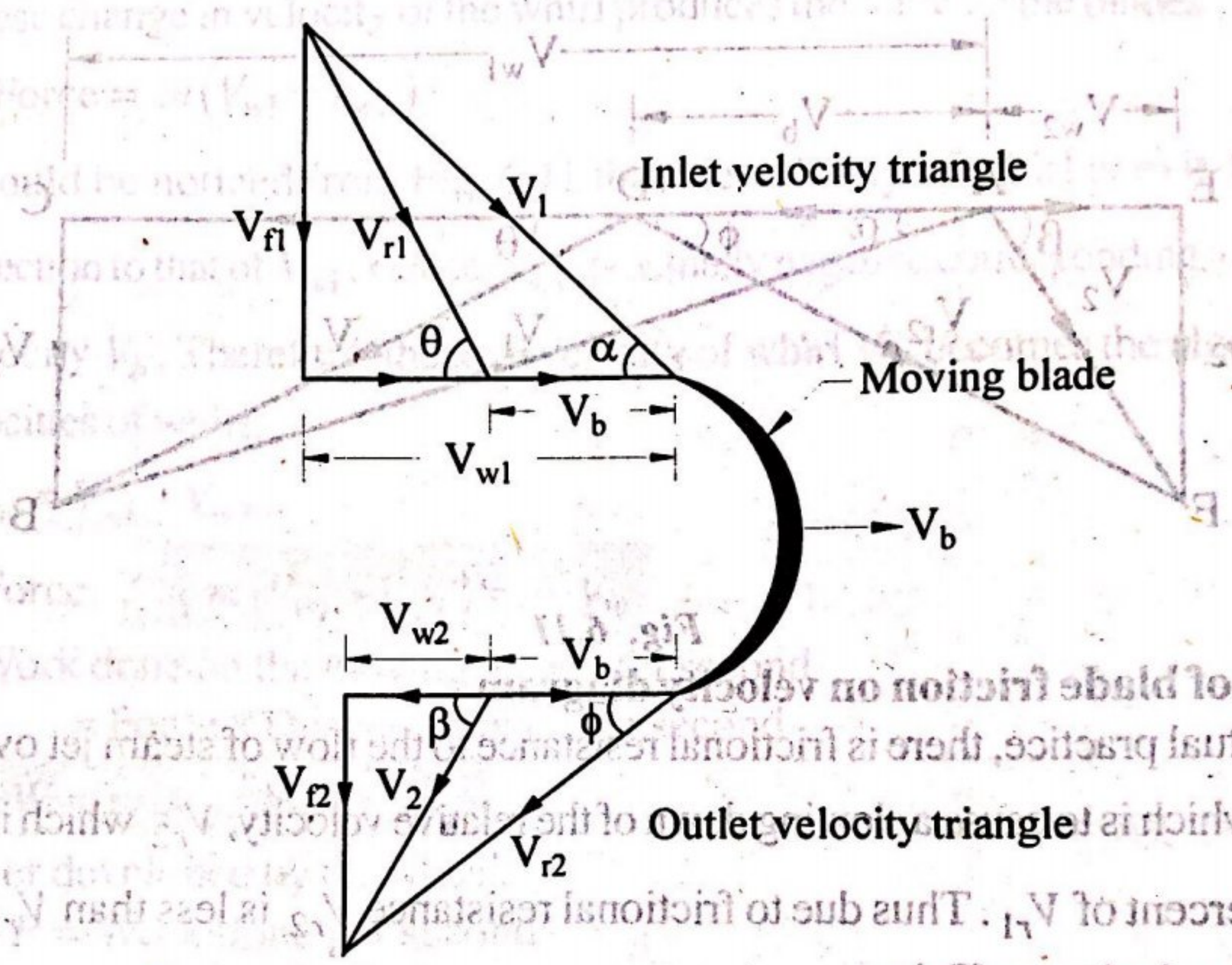


Fig. 6.10 Velocity diagram of an impulse turbine

Refer to Fig. 6.10

Let V_1 = absolute velocity of steam entering the moving blades, m/s.

V_2 = absolute velocity of steam at the outlet from the blade, m/s.

V_b = linear velocity of the moving blade, m/s.

V_{r1} = relative velocity of steam to moving blade at the inlet, m/s.

V_{r2} = relative velocity of steam to the moving blade at the outlet, m/s.

V_{f1} = Velocity of flow at inlet, m/s. It is the axial component of V_1.

V_{f2} = Velocity of flow at outlet, m/s. It is the axial component of V_2.

V_{w1} = Velocity of whirl at inlet, m/s. It is the horizontal component of V_1.

V_{w2} = Velocity of whirl at outlet, m/s. It is the horizontal component of V_2.

α = nozzle angle,

θ = blade angle at inlet,

ϕ = blade angle at outlet,

β = angle of discharge of the exit steam with the tangent to the wheel.

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Since both the inlet and the exit velocity triangles have the common vector, V_b, these two can be combined into a single diagram, for the sake of convenience in solving the problems. Assuming there is no frictional loss at the blades, relative velocity of steam while gliding over its surface will remain constant i.e. $V_{r1} = V_{r2}$. The combined velocity diagram for inlet and exit of moving blades is shown in Fig. 6.11

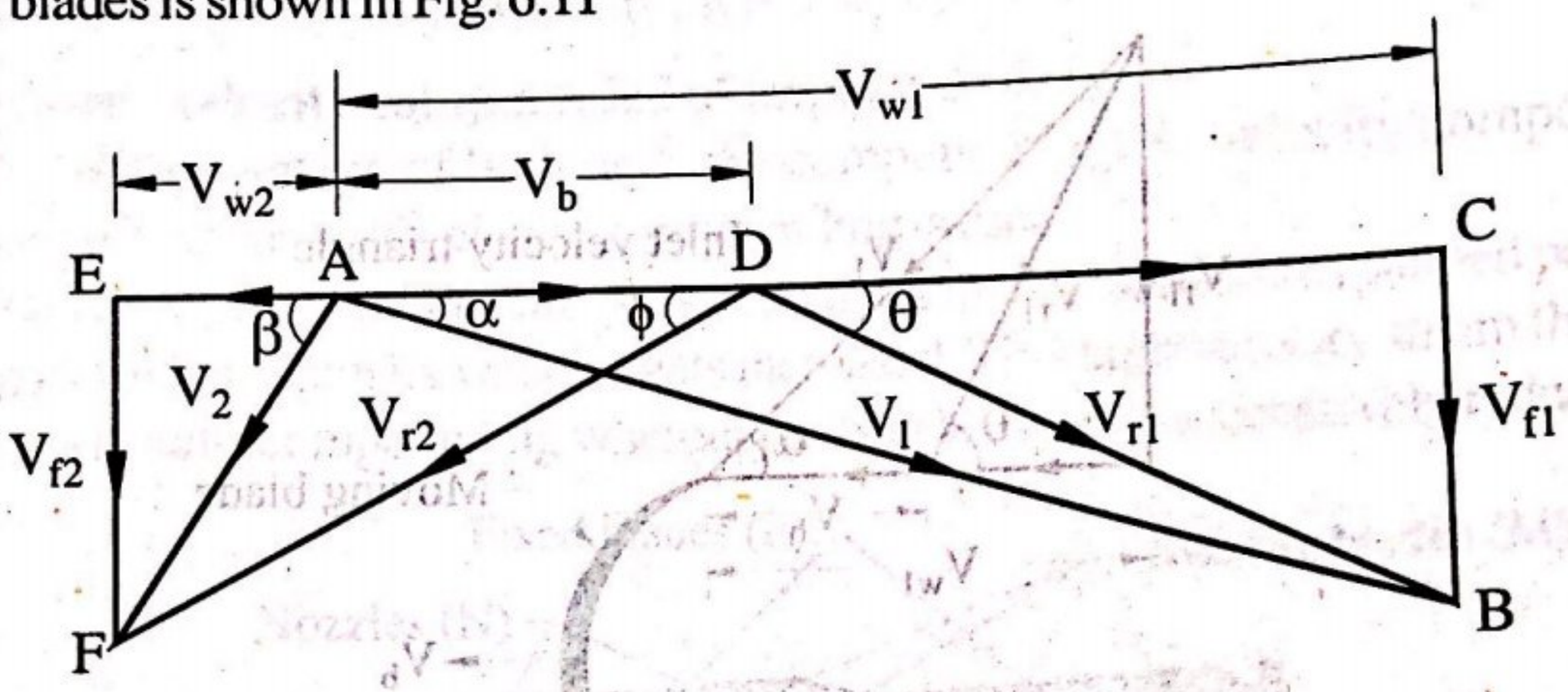


Fig. 6.11

- **The effect of blade friction on velocity diagram**

In actual practice, there is frictional resistance to the flow of steam jet over the blade, the effect of which is to cause a slowing down of the relative velocity, V_{r2} which is in the range of 85 to 90 percent of V_{r1}. Thus due to frictional resistance V_{r2} is less than V_{r1}. Therefore,

Blade velocity coefficient,

$$K = \frac{\text{Relative velocity at exit of moving blade, } V_{r2}}{\text{Relative velocity at inlet to moving blade, } V_{r1}}$$

$$\therefore V_{r2} = K V_{r1}$$

Fig. 6.12 shows velocity vector diagram considering effect of friction.

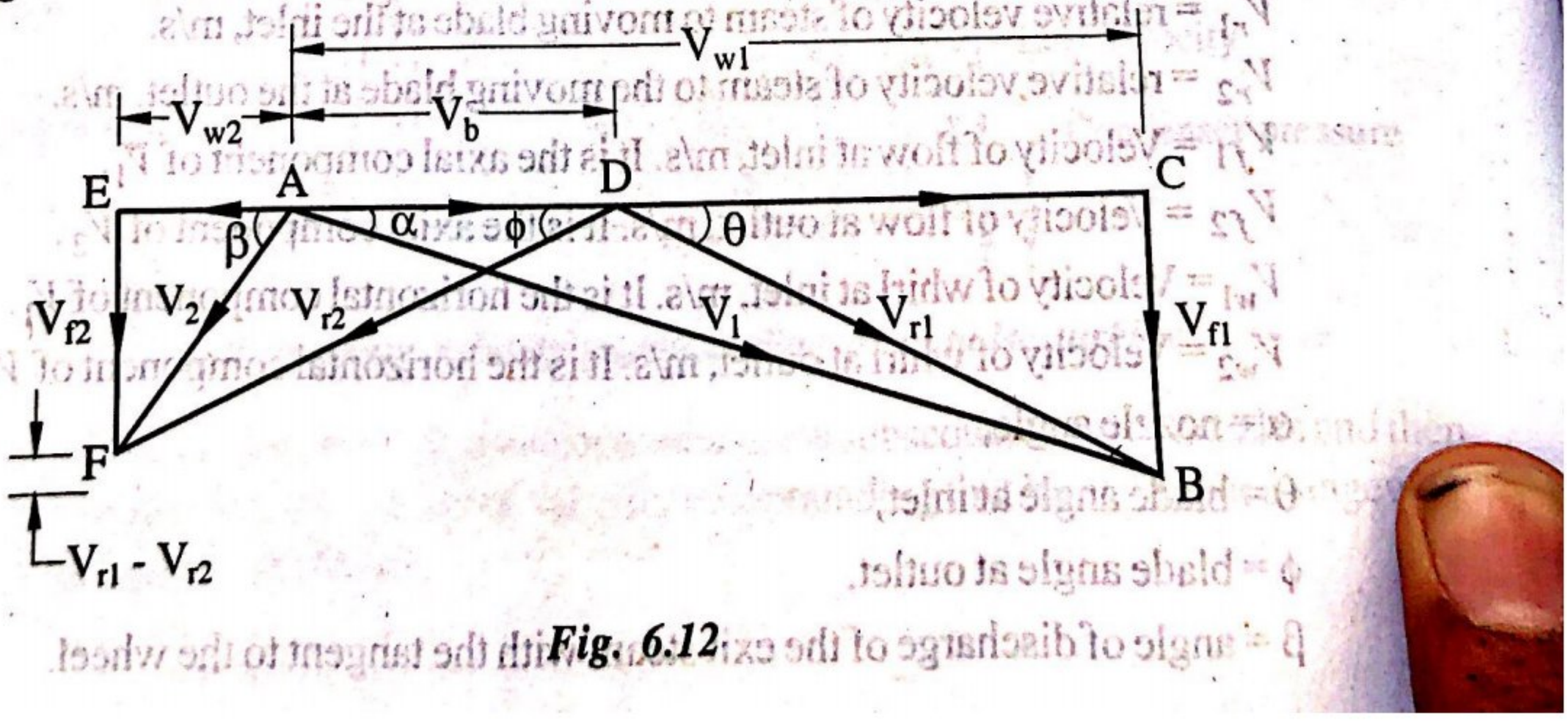


Fig. 6.12

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6.5.1 Calculations for work, power and efficiencies : [Sep. '13, May '13]

(1) Work and power :

By Newton's laws,

Force = Rate of change of momentum

= (Mass) × (Change of velocity)

These change in velocity of the whirl produces the force on the blades.

$$\therefore \text{Force} = m(V_{w1} - V_{w2})$$

It could be noticed from Fig. 6.11 that the velocity of whirl at exit V_{w2} is in the opposite direction to that of V_{w1}. Hence, V_{w2} is actually negative corresponding to the direction of blade velocity V_b. Therefore, the total velocity of whirl V_w becomes the algebraic sum of the two velocities of whirl.

$$\therefore V_w = V_{w1} + V_{w2}$$

$$\therefore \text{Force, } F = m(V_{w1} + V_{w2}) = m \cdot V_w \quad \ldots(6.1)$$

∴ Work done on the moving blades per second

= Force x Distance moved per second

$$W = m \cdot V_w \times V_b \quad J/s$$

Power developed by the wheel,

P = Work done per second

$$= m \cdot V_w \cdot V_b \text{ Watts} = \frac{mV_wV_b}{1000} \text{ kW} \quad \ldots(6.2)$$

(2) Blade efficiency :

It is also called ***diagram efficiency***. *It is the ratio of the workdone on the blades to the energy supplied to the blades.* The energy supplied per stage of an impulse turbine is equal to kinetic energy of steam exit from nozzle.

$$\therefore \eta_b = \frac{\textit{Work done on the blade}}{\textit{Energy supplied to the blade}}$$

$$= \frac{(m \cdot V_w \cdot V_b)}{\left(\frac{mV_1^2}{2}\right)} = \frac{2V_wV_b}{V_1^2} \quad \ldots(6.3)$$

Blade efficiency can also be defined as,

$$\eta_b = \frac{\textit{Change in K.E. of steam}}{\textit{K.E. supplied}}$$

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$$= \frac{m\left(\frac{V_1^2}{2} - \frac{V_2^2}{2}\right)}{\left(\frac{mV_1^2}{2}\right)} = \frac{V_1^2 - V_2^2}{V_1^2} \quad ..(6.4)$$

Blade efficiency will be maximum when V_2 is minimum i.e. when $\beta = 90°$ or the discharge is axial. **[Sep. '13]**

(3) Stage efficiency :

A row of nozzles (fixed blades) and a row of moving blades constitute a one stage of turbine. The energy supplied corresponds to the isentropic heat drop, ΔH in the nozzles. The stage efficiency η_s is given by

$$\therefore \eta_s = \frac{\textit{Work done}}{\textit{Energy supplied per stage}}$$

$$= \frac{m\, V_w V_b}{\Delta H \times 1000} = \frac{V_w\, V_b}{\Delta h \times 1000} \quad ...(6.5)$$

where Δh = isentropic heat drop per kg of steam, kJ/kg,

ΔH = isentropic heat drop, kJ $= m(\Delta h)$

The stage efficiency becomes equal to the blade efficiency if there are no friction losses in the nozzles.

$$\therefore \eta_s = \eta_n \times \eta_b$$

where η_n is efficiency of nozzle.

(4) Axial thrust : **[Sep. '13]**

Axial thrust, F_a = mass x acceleration in axial direction

= mass per second x change in velocity in axial direction.

$$F_a = m(V_{f1} - V_{f2}) \quad ...(6.6)$$

The axial thrust on the wheel is due to the force exerted as a result of the rate of change of momentum in axial direction. The thrust bearings are provided in the turbine in order to take the axial thrust on wheel.

6.6 Condition for maximum blade efficiency of impulse turbines

Let us consider the case of single stage impulse turbine. Hence, the effect of friction in the channel is taken into account and accordingly the velocity diagram is shown in Fig. 6.12. Due to friction V_{r2} is less than V_{r1} and $V_{r2} / V_{r1} = K$, where K is blade velocity coefficient,

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which is always less than unity.

We know that the blade efficiency is, $\eta_b = \dfrac{2\ V_w\ V_b}{V_1^2}$

From the velocity diagram find the value of $V_w\ V_b$

$$V_w = V_{w1} + V_{w2} = CD + DE$$

$$= V_{r1}\cos\theta + V_{r2}\cos\phi = V_{r1}\cos\theta\left(1 + \frac{V_{r2}\cos\phi}{V_{r1}\cos\theta}\right)$$

$$= V_{r1}\cos\theta\,(1 + KB) \text{ where } B = \frac{\cos\phi}{\cos\theta}$$

Since θ and ϕ are usually almost equal, so for this analysis we assume that B is a constant, We have

$$V_{r1}\cos\theta = V_1\cos\alpha - V_b$$

$$\therefore V_w = (V_1\cos\alpha - V_b)(1 + KB)$$

$$\therefore 2V_w V_b = 2V_b\,(V_1\cos\alpha - V_b)(1 + KB)$$

$$\therefore \eta_b = \frac{2V_b\,(V_1\cos\alpha - V_b)(1 + KB)}{V_1^2}$$

$$\therefore \eta_b = 2\left(\rho\cos\alpha - \rho^2\right)(1 + KB) \quad \ldots(6.7)$$

where $\rho = \dfrac{V_b}{V_1}$ *is the ratio of blade speed to steam speed and is commonly called as* ***Blade speed ratio***. The blade speed ratio is a very important factor in the design of steam turbines. The efficiency of the turbine depends upon this factor.

Equation (6.7) is the equation of the parabola which is shown in Fig. 6.13. From the graph is obvious that the blade efficiency increases as ρ is increased until ρ reaches a value which is approximately one-half, after which blade efficiency decreases.

For maximum blade efficiency, differentiation with respect ρ should be zero.

$$\therefore \frac{d\eta_b}{d\rho} = 0 = 2(\cos\alpha - 2\rho)(1 + KB)$$

$$\therefore 2(1 + KB)(\cos\alpha - 2\rho) = 0$$

$$\therefore \cos\alpha - 2\rho = 0$$

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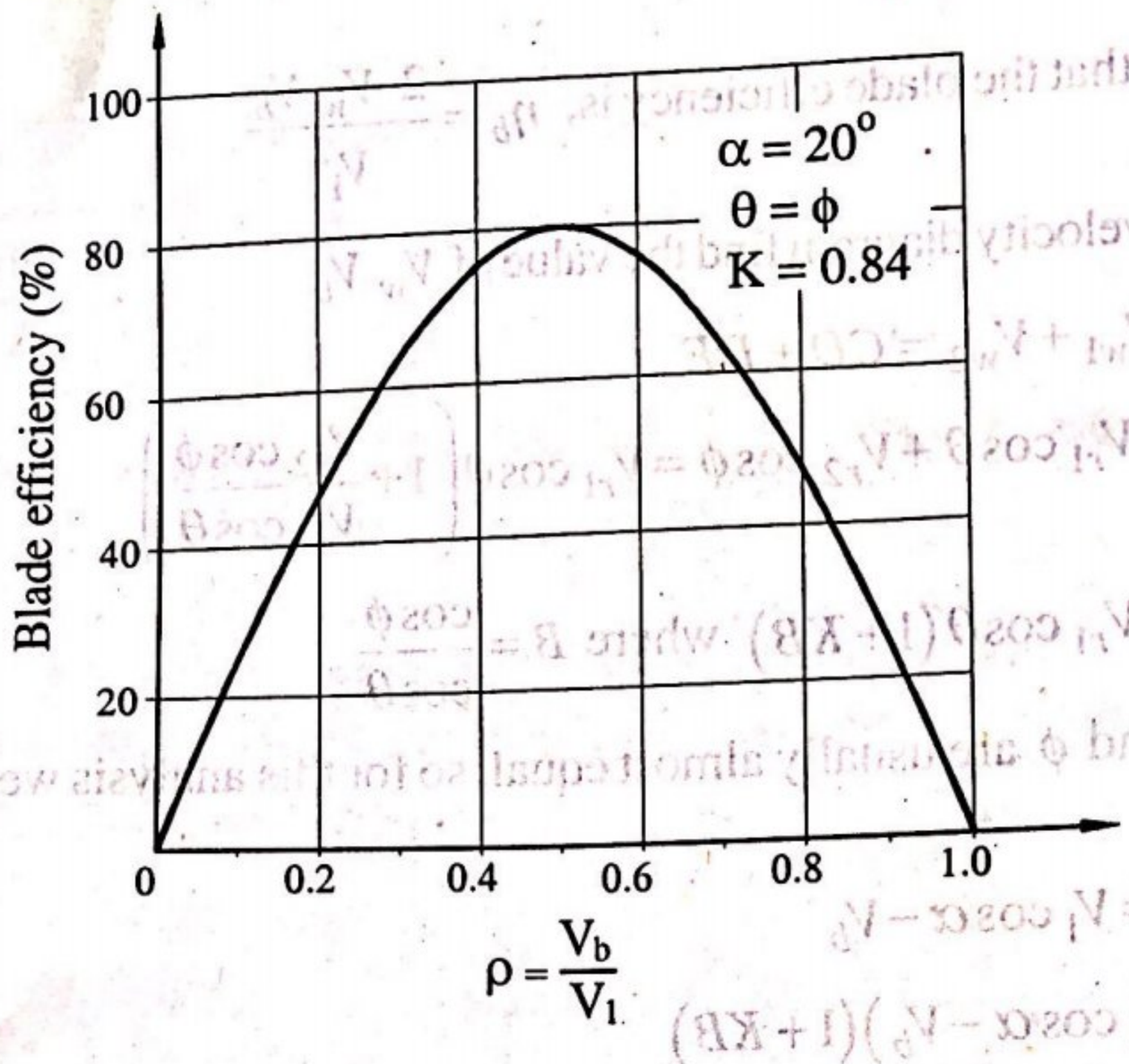


Fig. 6.13 Blade efficiency

Therefore, optimum value of ρ is

$$\rho_{opt} = \left(\frac{V_b}{V_1}\right)_{opt} = \frac{\cos\alpha}{2} \qquad ...(6.8)$$

Substituting this value of ρ in equation (6.7), we get

$$(\eta_b)_{max} = 2(1+KB)\left(\frac{\cos\alpha\cos\alpha}{2} - \frac{\cos^2\alpha}{4}\right)$$

$$= (1+KB)\frac{\cos^2\alpha}{2}$$

It is sufficiently accurate to assume symmetrical blades $(\theta = \phi)$ and no friction in fluid passage for the purpose of analysis.

$\therefore K = 1$, and $B = 1$

$$\therefore (\eta_b)_{max} = \cos^2\alpha \qquad ...(6.9)$$

The maximum power output per kg of steam from a single stage impulse turbine having equiangular blades and for without friction is given by :

$$\text{Maximum Power, } (P)_{max} = (V_w V_b)_{max}$$

$$= \left[V_b (V_1\cos\alpha - V_b)(1+KB)\right]_{max} \quad W/kg$$

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Substituting the value of $\left(\frac{V_b}{V_1}\right)_{opt}$, we have

$$P_{max} = 2V_b^2 \quad \text{for} \quad K = 1 \text{ and } B = 1 \qquad ...(6.10)$$

Problem 6.1 *Steam issues from the nozzles of a De Laval turbine with a velocity of 1400 m/s. The nozzle angle is 15°, the mean blade velocity is 450 m/s and the inlet and outlet angle of blade are equal. The mass of steam flowing through turbine per hour is 1000 kg. Determine : (i) the blade angles, (ii) the relative velocity of steam entering the blades, (iii) the tangential force on the blades, (iv) the power developed, (v) the blade efficiency. Assume that $K = 0.8$.* ***[May '15, same as Nov. '14]***

Solution : *Given data*

$V_1 = 1400\ m/s,\ \alpha = 15°,\ V_b = 450\ m/s,\ \theta = \phi,\ m = 1000\ kg/h = 1000/3600\ kg/s,$

$K = V_{r2}/V_{r1} = 0.8$

(i) The blade angles,

With this data, the velocity diagram is shown in Fig. 6.14. The blade angle at inlet θ is measured which gives $\theta = 22°$.

Since the blades are symmetrical, so

$$\theta = \phi = \mathbf{22°} \qquad \textbf{Ans}$$

(ii) The relative velocity of steam entering the blades.

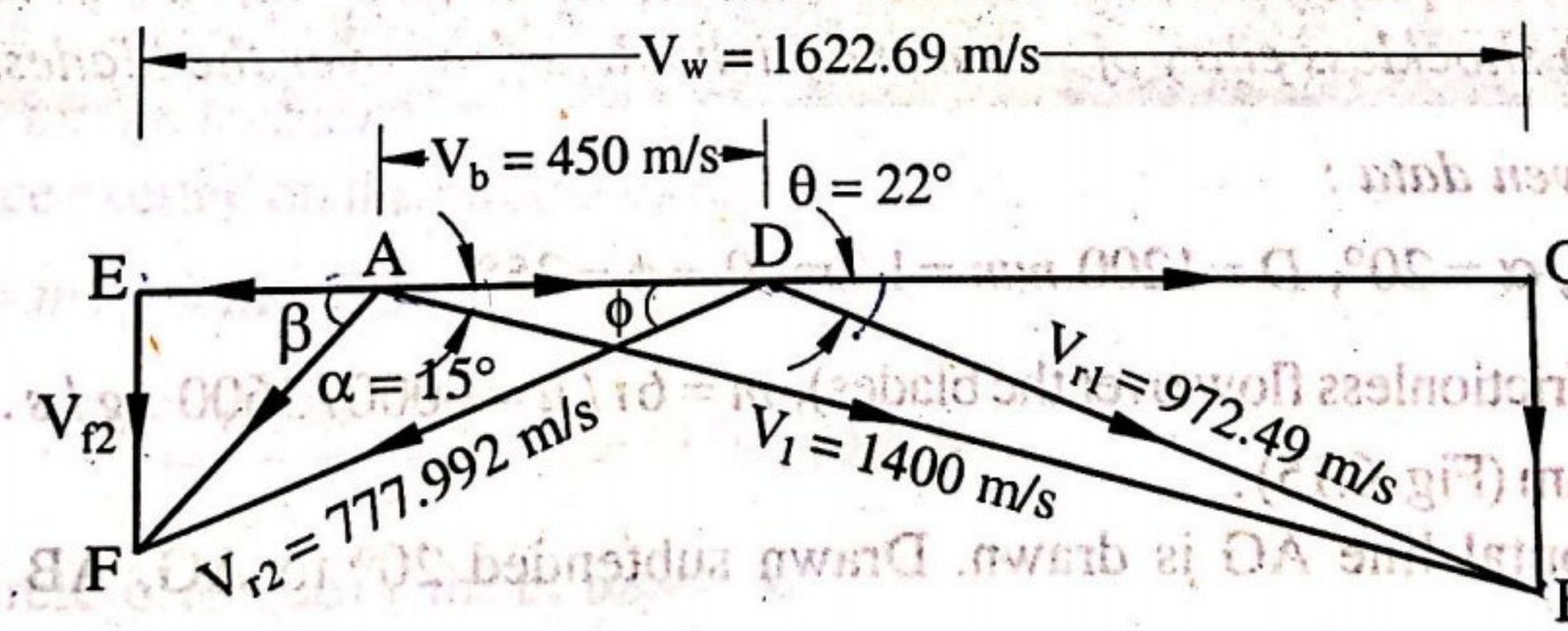


Fig. 6.14

The relative velocity at inlet V_{r1} is measured which gives

$$V_{r1} = \mathbf{972.49\ m/s} \qquad \textbf{Ans}$$

The relative velocity at outlet, $V_{r2} = KV_{r1} = 0.8 \times 972.49$

$$= 777.992\ m/s$$

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and $\phi = 22°$, so the outlet velocity triangle can be drawn.

$$\therefore V_w = V_{w1} + V_{w2} = 1622.69 \ m/s$$

(iii) The tangential force on the blades,

$$F = m(V_{w1} + V_{w2}) = mV_w$$

$$= \frac{1000}{3600} \times 1622.69 = \mathbf{450.747\,N} \qquad \textit{Ans}$$

(iv) The power developed,

$$P = \frac{mV_wV_b}{1000} = \frac{1000}{3600} \times \frac{1622.69 \times 450}{1000} \ kW$$

$$= \mathbf{202.836\ kW} \qquad \textit{Ans}$$

(v) The blade efficiency,

$$\eta_b = \frac{2V_bV_w}{V_1^2} = \frac{2 \times 450 \times 1622.69}{(1400)^2} = 0.7451$$

$$= \mathbf{74.51\ \%} \qquad \textit{Ans.}$$

Problem 6.2 *Steam with a velocity of 700 m/s. is directed upon the blades of a single stage impulse turbine at an angle 20° to the direction of motion. The average diameter of the blade ring is 1200 mm. The blade angles at inlet and exit are the same and equal to 35°. Determine (i) the speed of the turbine, (ii) the absolute velocity of steam leaving the blades, (iii) torque on the turbine rotor when the steam flow rate is 6 t/h. Base your calculation on shockless entry of steam and frictionless flow over the blades.*

Solution : *Given data :*

$V_1 = 700 \ m/s, \alpha = 20°, D = 1200 \ mm = 1.2m, \theta = \phi = 35°,$

$V_{r1} = V_{r2}$ (∵ frictionless flow over the blades), $m = 6\,t/h = 6000/3600 \ kg/s$.

Velocity diagram (Fig. 6.15) :

Horizontal line AG is drawn. Drawn subtended 20° to AG, AB represents $V_1 = 700 \ m/s$.

Since it is known that inlet angle $(\angle GDB)$ will be $\theta = 35°$,

$$\angle ADB = 180 - 35° = 145°$$

$$\therefore \angle ABD = 180 - (20+145) = 15°$$

Hence from B, BD is drawn making 15° angle with AB.

(i) The speed of the turbine,

From the velocity diagram,

AD = blade speed = V_b = 319.29 m/s

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We have, blade speed $V_b = \frac{\pi DN}{60}$

$$\therefore 319.29 = \frac{\pi \times 1.2 \times N}{60}$$

$\therefore$ N = **5081.658 rpm** ***Ans***

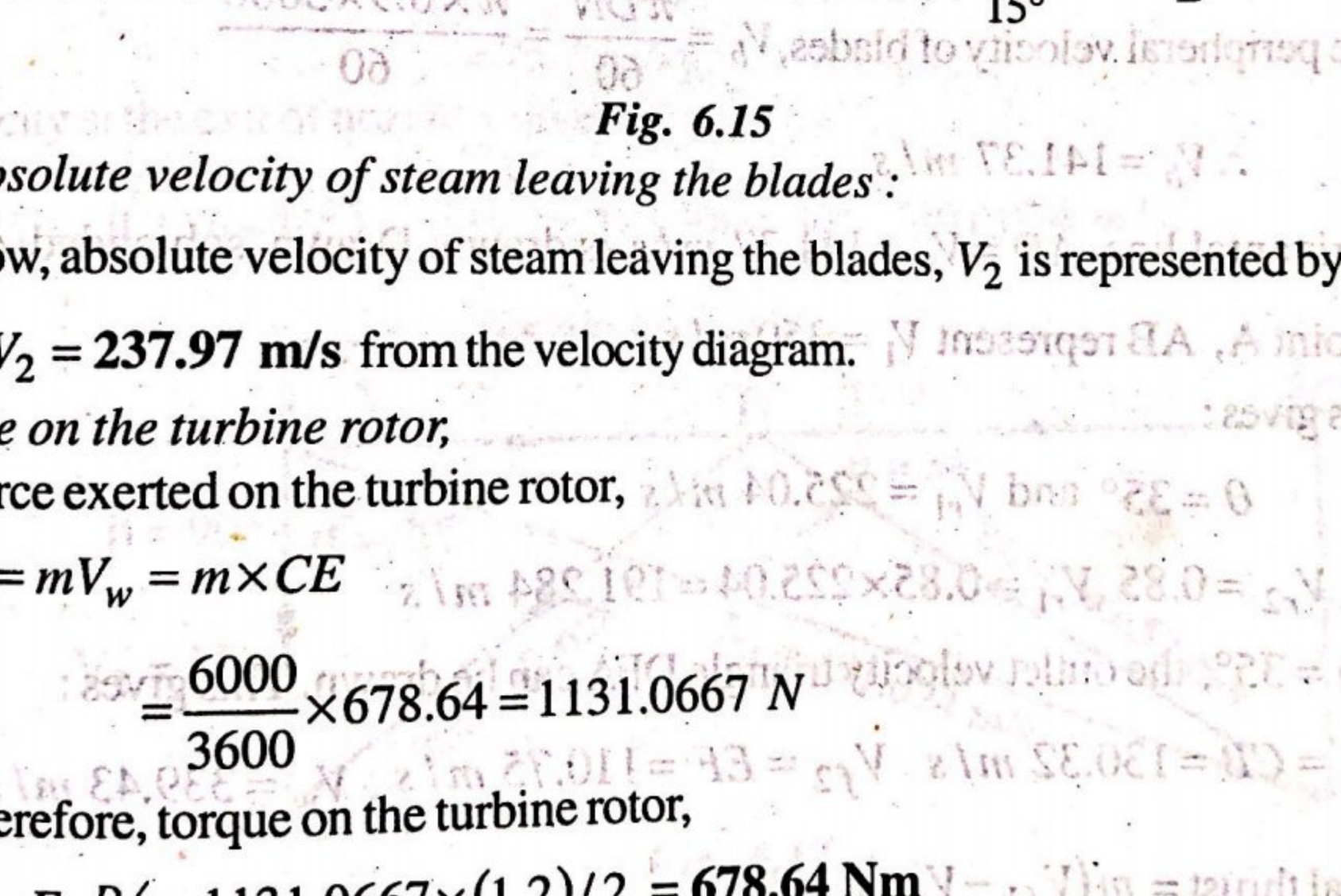


Fig. 6.15

(ii) The absolute velocity of steam leaving the blades :

Now, absolute velocity of steam leaving the blades, V_2 is represented by AF.

$\therefore V_2$ = **237.97 m/s** from the velocity diagram. ***Ans***

(iii) Torque on the turbine rotor,

Force exerted on the turbine rotor,

$$F = mV_w = m \times CE$$

$$= \frac{6000}{3600} \times 678.64 = 1131.0667\ N$$

Therefore, torque on the turbine rotor,

$$T = F \cdot {}^{D}/_{2} = 1131.0667 \times (1.2)/2 = \mathbf{678.64\ Nm}$$ ***Ans***

Problem 6.3 *In a stage of an impulse turbine provided with a single row wheel, the mean diameter of the blade ring is 90 cm and the speed of rotation is 3,000 rpm. The steam issues from the nozzles with a velocity of 350 m/s and the nozzle angle is 22°. The rotor blades are equiangular and due to friction in the blade channels and relative velocity of steam at outlet from the blades is 0.85 times the relative velocity of the steam entering the blades. What is the power developed in the blades when the axial thrust on the blades is 150 N?*

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Solution : *Given data :*

$D = 90\,cm = 0.9\,m,\ \alpha = 22°,\ N = 3000\ rpm,\ V_1 = 350\ m/s,\ \theta = \phi,\ V_{r2} = 0.85 V_{r1},$

Axial thrust = 150 N, P = ?

Velocity diagram is shown in Fig. 6.16.

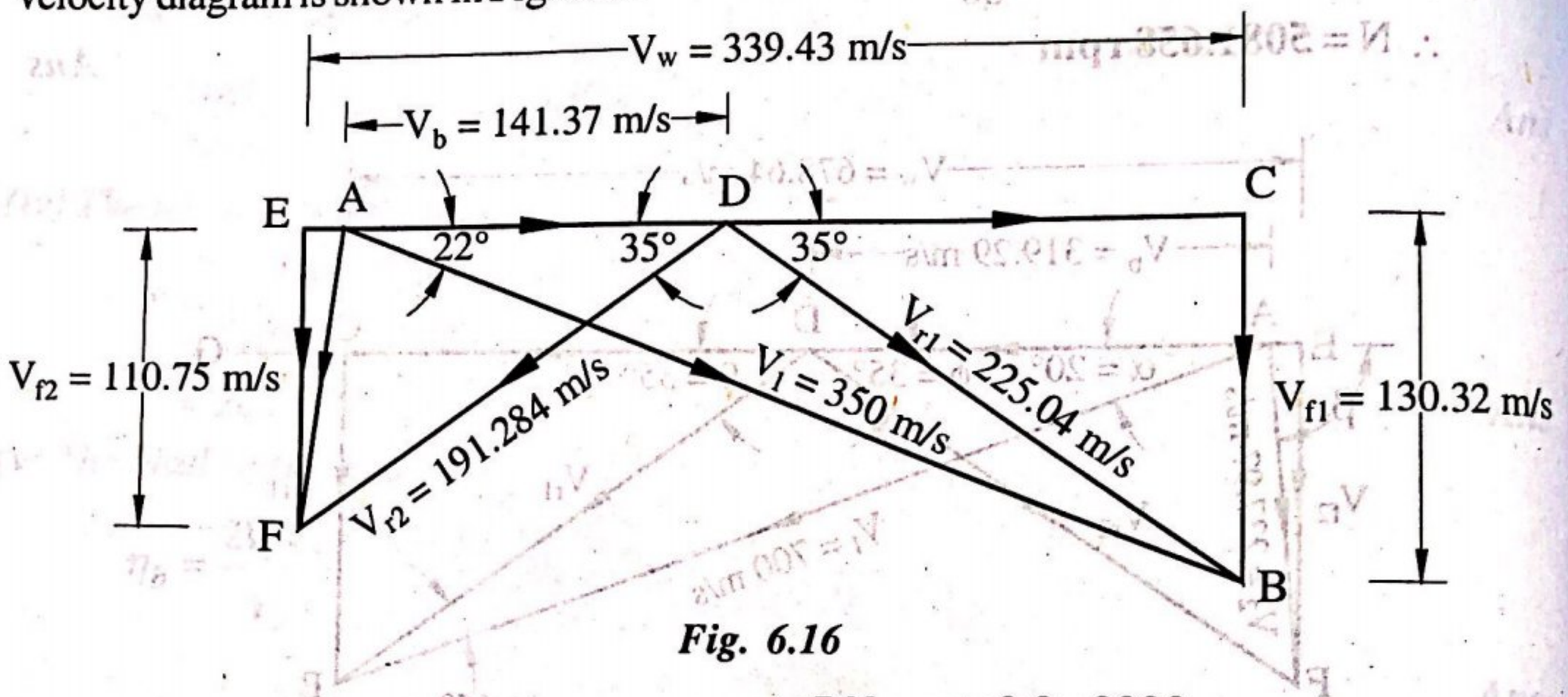


Fig. 6.16

The peripheral velocity of blades, $V_b = \dfrac{\pi DN}{60} = \dfrac{\pi \times 0.9 \times 3000}{60}$

$\therefore V_b = 141.37\ m/s$.

Horizontal line $AD = V_b = 141.37\ m/s$ is drawn. Drawn subtended $\alpha = 22°$ to AD from point A, AB represent $V_1 = 350\,m/s$

This gives :

$\theta = 35°$ and $V_{r1} = 225.04\ m/s$

So, $V_{r2} = 0.85\ V_{r1} = 0.85 \times 225.04 = 191.284\ m/s$

Since $\theta = \phi = 35°$, the outlet velocity triangle DFA can be drawn. This gives :

$V_{f1} = CB = 130.32\ m/s\ \ V_{f2} = EF = 110.75\ m/s$, $V_w = 339.43\ m/s$.

Axial thrust $= m\left(V_{f1} - V_{f2}\right)$

$\therefore 150 = m(130.32 - 110.75)$

$\therefore m = 7.664\ kg/s$

Power developed, $P = \dfrac{m\,V_w V_b}{1000} = \dfrac{7.664 \times 339.43 \times 141.37}{1000}$

$= \mathbf{367.76\ kW}$

Ans

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Problem 6.4 *Steam at a pressure of 3 bar with a velocity of 50 m/s and the dryness fraction of 0.95 reaches the nozzles of one stage of an impulse turbine. The pressure of steam that drops along the nozzles is 1.1 bar and the 6% of the isentropic enthalpy drop is absorbed in overcoming friction. The nozzles are inclined at an angle of 20° to the plane of wheel, and the ratio of blade velocity to the tangential component of steam velocity is 0.45. If the blade velocity coefficient is 0.85, find the blade angles so that the steam may enter the blade without shock and leave them in an axial direction. Find also the power input and frictional losses over the blades per kg of steam.*

Solution : Given data

Let 0 and 1 represent the inlet and outlet of nozzle

$p_o = 3\ bar,\ V_o = 50\ m/s,\ x_o = 0.95,\ p_1 = 3 - 1.1 = 1.9\ bar,$ nozzle efficiency,

$\eta_n = (1 - 0.06) = 0.94,\ \alpha = 20°,\ \dfrac{V_b}{V_1 \cos\alpha} = 0.45,\ K = 0.85, \theta = ?, P_i = ?,$

Friction losses = ?.

From Mollier's diagram the isentropic heat drop per kg of steam is calculated as

$$(\Delta h)_{isen} = 80\ kJ/kg$$

Actual heat drop, $\Delta h = \eta_n \times (\Delta h)_{isen} = 0.94 \times 80$

$= 75.2$ kJ/kg.

The velocity at the exit of nozzle is given by

$$V_1 = \sqrt{(2\Delta h + V_0^2)} = \sqrt{2 \times 75.2 \times 1000 + 50^2} = 391.024\ m/s$$

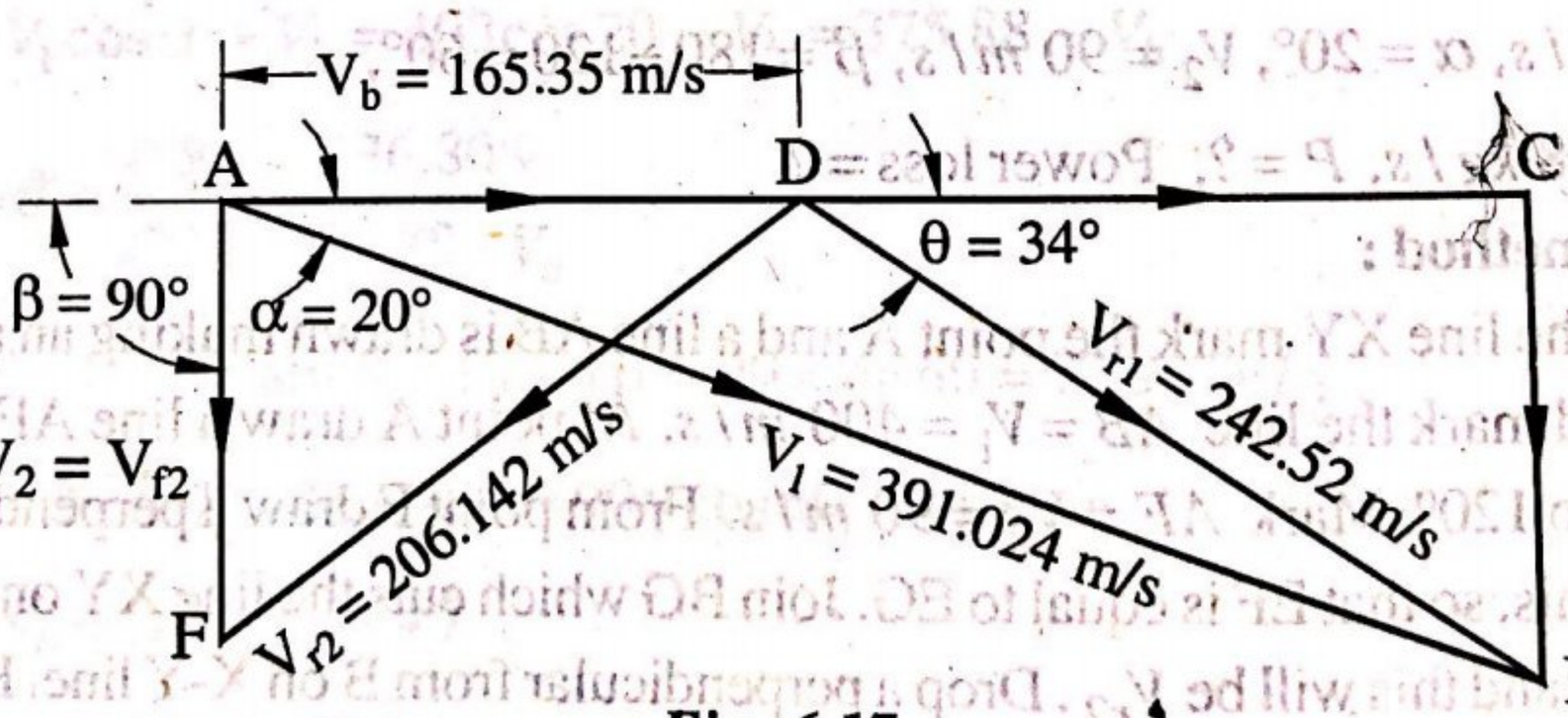


Fig. 6.17

Since $\dfrac{V_b}{V_1 \cos\alpha} = 0.45,$

$\therefore V_b = 0.45 \times 391.024 \times \cos 20,\quad \therefore V_b = 165.35\ m/s$

Velocity diagram is shown in Fig. 6.17

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With the known values of V_b, V_1 and α the inlet velocity triangle ABD can be drawn.

This gives $\theta = 34°$, $V_{r1} = 242.52\ m/s$.

$$V_{r2} = KV_{r1} = 0.85 \times 242.52 = 206.142\ m/s$$

Since the steam leaves axially i.e. $\beta = 90°$, DF can be intercepted on the line AF with the centre at D and radius equal to $V_{r2} = 206.142\ m/s$.

Power input, P_i = K.E. of steam leaving the nozzle

$$= \frac{V_1^2}{2} = \frac{(391.024)^2}{2} = 76449.88\ J/kg \cong \mathbf{76.45\ kJ/kg} \quad \textit{Ans}$$

$$\text{Blade friction losses} = \frac{V_{r1}^2 - V_{r2}^2}{2 \times 1000}\ kJ/kg$$

$$= \frac{(242.52)^2 - (206.142)^2}{2 \times 1000} = \mathbf{8.1607}\ kJ/kg \quad \textit{Ans}$$

Problem 6.5 *Steam flows from the nozzles of a single row impulse turbine with a velocity 400 m/s at a direction which is inclined at an angle of 20° to the peripheral velocity. Steam comes out of the moving blades with an absolute velocity of 90 m/s in the direction at 120° with the direction of blade motion. The blades are equiangular and steam flow rate is 8 kg/s. Calculate the power loss due to friction.*

Solution : Given data

$V_1 = 400\ m/s$, $\alpha = 20°$, $V_2 = 90\ m/s$, $\beta = 180 - 120 = 60°$,

$\theta = \phi$, $m = 8\ kg/s$, $P = ?$, Power loss = ?

Graphical method :

On the line XY mark the point A and a line AB is drawn making an angle $\alpha = 20°$ with XY and mark the line $AB = V_1 = 400\ m/s$. At point A draw a line AF making angle XAF equal to 120°. Mark $AF = V_2 = 90\ m/s$. From point F draw a perpendicular line XY and extend this, so that EF is equal to EG. Join BG which cuts the line XY on D. DB will be V_{r1}. Join DF and this will be V_{r2}. Drop a perpendicular from B on X-Y line. Hence triangles DEG, DEF and BCD are similar. Thus $\theta = \phi$.

This diagram gives $V_b = 105.10\ m/s$, $V_{r1} = 303.74\ m/s$, $V_{r2} = 169.27\ m/s$, $V_w = 420.09\ m/s$.

$$\therefore \text{ Power developed, } P = \frac{m\, V_w\, V_b}{1000} = \frac{8 \times 420.09 \times 105.10}{1000} = \mathbf{353.21\ kW} \quad \textit{Ans}$$

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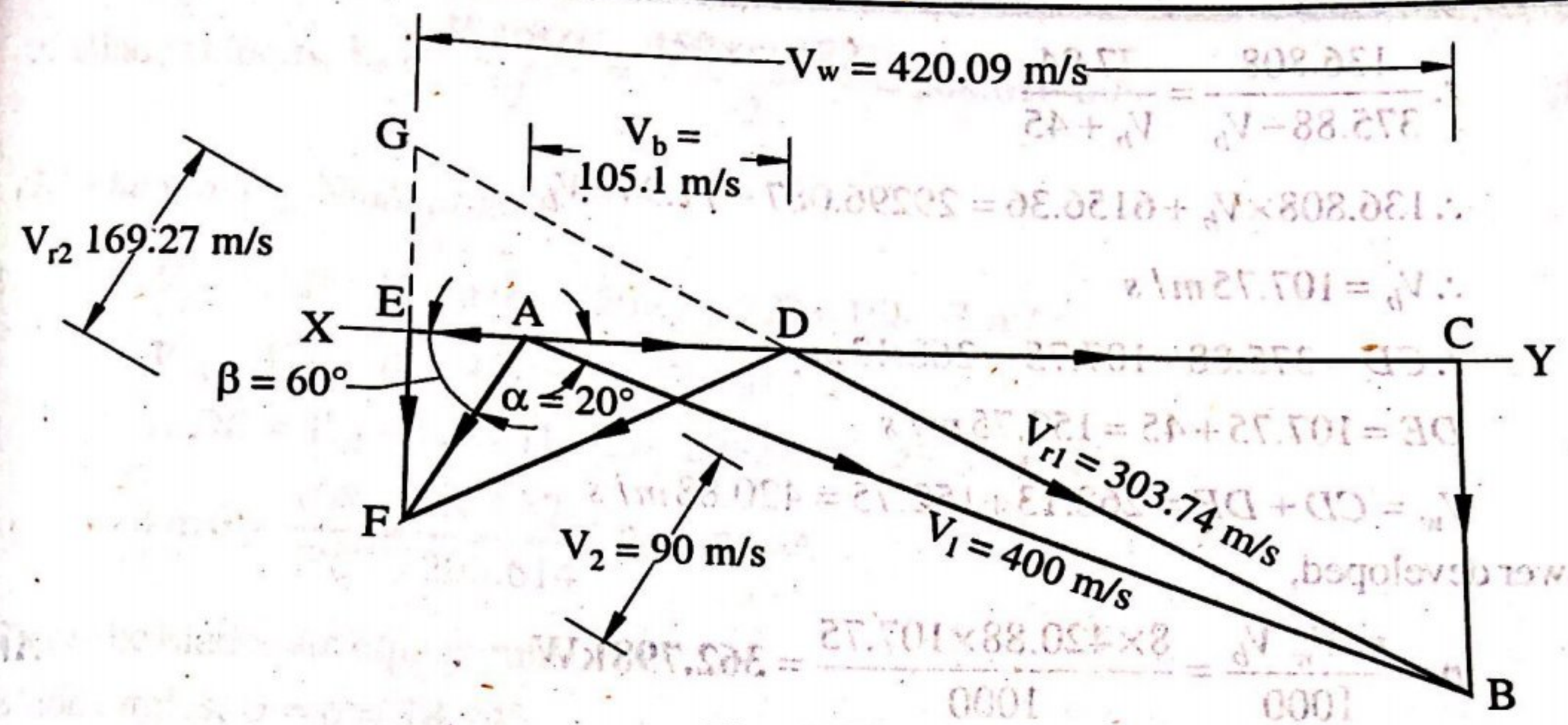


Fig. 6.18

$$\text{Power loss due to friction} = \frac{m\left(V_{r1}^2 - V_{r2}^2\right)}{2\times 1000}$$

$$= \frac{8\left((303.74)^2 - (169.27)^2\right)}{2\times 1000}$$

$$= \mathbf{254.42\ kW} \qquad \textit{Ans}$$

Analytical method :

$$CB = V_1 \sin\alpha = 400 \sin 20 = 136.808\ m/s$$

$$CD = V_1 \cos\alpha - V_b = 400\cos 20 - V_b = 375.88 - V_b$$

$$\therefore\ \tan\theta = \frac{CB}{CD} = \frac{136.808}{375.88 - V_b}$$

$$EF = V_{f2} = AF \sin\beta = V_2 \sin\beta = 90\times\sin 60 = 77.94\ m/s$$

$$AE = AF\cos\beta = V_2\cos\beta = 90\times\cos 60 = 45\ m/s$$

$$DE = V_b + AE = V_b + 45$$

$$\therefore\ \tan\phi = \frac{EF}{DE} = \frac{77.94}{V_b + 45}$$

since $\phi = \theta$

$\therefore \tan\theta = \tan\phi$

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$$\therefore \frac{136.808}{375.88 - V_b} = \frac{77.94}{V_b + 45}$$

$$\therefore 136.808 \times V_b + 6156.36 = 29296.087 - 77.94 \times V_b$$

$$\therefore V_b = 107.75\, m/s$$

$$\therefore CD = 375.88 - 107.75 = 268.13\, m/s$$

$$DE = 107.75 + 45 = 152.75\, m/s$$

$$V_w = CD + DE = 268.13 + 152.75 = 420.88\, m/s$$

Power developed,

$$p = \frac{m\, V_w\, V_b}{1000} = \frac{8 \times 420.88 \times 107.75}{1000} = \mathbf{362.798 kW} \qquad \textit{Ans}$$

$$V_{r1} = \sqrt{(CB)^2 + (CD)^2} = \sqrt{(136.808)^2 + (268.13)^2}$$
$$= 301.015\ m/s$$

$$V_{r2} = \sqrt{(DE)^2 + (EF)^2} = \sqrt{(152.75)^2 + (77.94)^2}$$
$$= 171.48\, m/s$$

$$\therefore \text{Power loss due to friction} = \frac{m\left(V_{r1}^2 - V_{r2}^2\right)}{2 \times 1000}$$

$$= \frac{8\left[(301.015)^2 - (171.48)^2\right]}{2 \times 1000} = \mathbf{244.818\ kW} \qquad \textit{Ans}$$

Problem 6.6 *Steam issues from the nozzles at angle of 22^0 at a velocity of 450 m/s. The friction factor is 0.86. For a single stage turbine designed for maximum efficiency, Calculate :*
(i) the blade velocity, (ii) the moving blade angles for equiangular blades, (iii) the blade efficiency, (iv) the stage efficiency if the nozzle efficiency is 94%. (v) Power developed for a mass flow rate of steam of 5 kg/s.

Solution:- Given data: (Refer Fig. 6.12)

$$\alpha = 22°, V_1 = 450\ m/s, K = 0.86, \eta_n = 94\%, m = 5 kg/s$$

(i) Blade velocity,

For maximum efficiency blade speed ratio is given by

$$\rho = \frac{\cos\alpha}{2} = \frac{V_b}{V_1}$$

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$\therefore$ Blade velocity, $V_b = \dfrac{V_1 \cos\alpha}{2} = \dfrac{450 \times \cos 22}{2} = \mathbf{208.616\ m/s}$ ***Ans***

(ii) The moving blade angles:

$$V_{f1} = CB = V_1 \sin\alpha = 450 \times \sin 22° = 168.57\ m/s$$

$$V_{w1} = V_1 \cos\alpha = 450 \times \cos 22 = 417.23\ m/s$$

$$\therefore DC = V_{w1} - V_b = 417.23 - 208.616 = 208.614\ m/s$$

$$\tan\theta = \frac{CB}{DC} = \frac{168.57}{208.614};\ \theta = 38.94°$$

Since the blades are equiangular,

Blade angles, $\theta = \phi = \mathbf{38.94°}$ ***Ans***

(iii) Blade efficiency :

$$(\eta_b)_{max} = \cos^2\alpha = \cos^2 22 = 0.8597 = \mathbf{85.97\%}$$ ***Ans***

(iv) Stage efficiency = Blade efficiency $\times$ Nozzle efficiency

$= 0.8597 \times 0.94 = 0.8081 = \mathbf{80.81\%}$ ***Ans***

(v) Power developed :

Maximum work done per kg of steam

$$= V_b (V_1 \cos\alpha - V_b)(1 + KB)$$

$$= 208.616(450 \cos 22 - 208.616)(1 + 0.86 \times 1)$$

$$= 80948.667\ W/kg$$

Power developed,

$$P = \frac{m \times Work\ done/kg\ of\ steam}{1000} = \frac{5 \times 80948.667}{1000} = \mathbf{404.743 kW}$$ ***Ans***

Problem 6.7 *A single stage impulse turbine has a mean blade ring diameter of 120 cm and runs at 3500 rpm. The blade speed ratio is 0.45 and discharge is axial. The nozzle angle is 16^0 and blade friction factor is 0.9. Determine : (i) blade angles, and (ii) theoretical specific power output.* **[Same as May '12]**

Solution: Given data: (Refer Fig. 6.19)

$D = 120\ cm = 1.2\ m$, $N = 3500\ rpm$, $\rho = \left(\dfrac{V_b}{V_1}\right) = 0.45$, $\beta = 90°$ *(discharge is axial)*,

$\alpha = 16°$; $K = 0.9$

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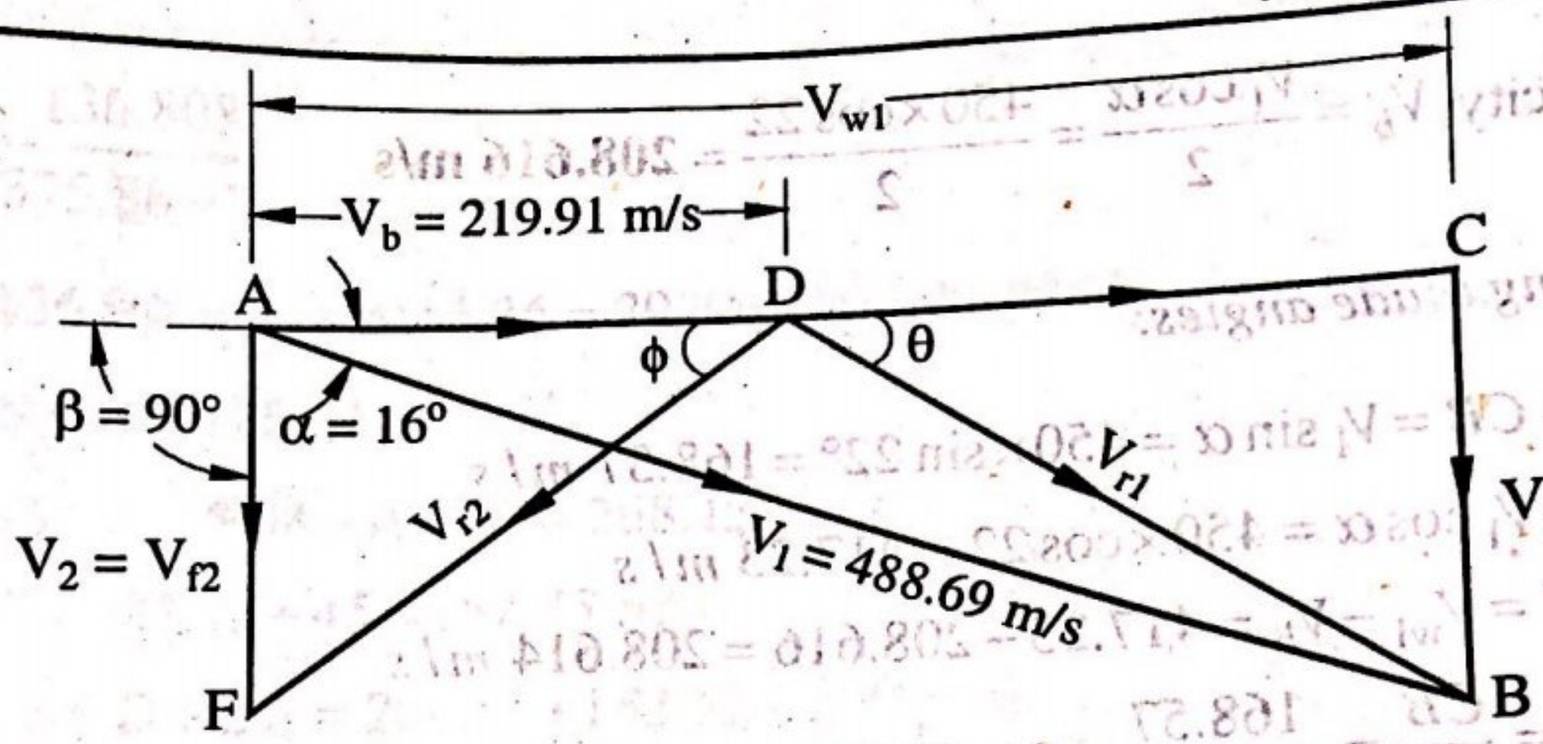


Fig. 6.19

(i) Blade angles:

Blade Velocity, $V_b = \dfrac{\pi DN}{60} = \dfrac{\pi \times 1.2 \times 3500}{60} = 219.91\ m/s$

$$V_1 = \frac{V_b}{\rho} = \frac{219.91}{0.45} = 488.69\ m/s$$

From the velocity diagram,

$$V_{w1} = V_1 \cos\alpha = 488.69 \cos 16 = 469.759\ m/s$$

$$\therefore DC = V_{w1} - V_b = 469.759 - 219.91 = 249.849\ m/s$$

$$BC = V_{f1} = V_1 \sin\alpha = 488.69 \sin 16° = 134.70\ m/s$$

$$V_{r1} = \sqrt{(DC)^2 + (BC)^2} = \sqrt{(249.849)^2 + (134.70)^2} = 283.846\ m/s$$

$$\tan\theta = \frac{BC}{DC};\ \theta = \tan^{-1}\left(\frac{134.70}{249.849}\right) = 28.33°$$

∴ Inlet angle of moving blade, $\theta = \mathbf{28.33°}$ ***Ans***

$$V_{r2} = KV_{r1} = 0.9 \times 283.846 = 255.46\ m/s$$

$$\cos\phi = \frac{V_b}{V_{r2}};\ \phi = \cos^{-1}\left(\frac{219.91}{255.46}\right) = 30.59°$$

∴ Exit angle of moving blade, $\phi = \mathbf{30.59°}$ ***Ans***

(ii) Specific power output:

$V_{w2} = 0$ $(\because$ *Discharge is axial*$)$

For specific power output, m = 1 kg of steam/s

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$$P = \frac{m(V_{w1} + V_{w2})V_b}{1000} \ kW$$

$$= \frac{1 \times (469.759 + 0) \times 219.91}{1000}$$

$$= \mathbf{103.30\ kW\ per\ kg\ of\ steam}$$ **Ans**

This problem can be solved by graphical method also.

Problem 6.8 *A simple impulse turbine has a mean blade ring diameter of 100 cm and runs at 5000 rpm. The nozzle angle is 16^0 and the steam leaves at nozzles with a velocity of 800 m/s. The blades are equiangular and the blade friction factor is 0.9. Determine : (i) the inlet angles of the blade for shockless entry of steam, (ii) the diagram power for a steam flow of 800 kg/h (iii) the diagram efficiency, (iv) the axial thrust, and (v) the loss of K.E due to friction.* ***[May '15, same as May '13]***

Solution: Given data: (Refer Fig. 6.20)

D = 100 cm = 1 m, N = 5000 rpm, $\alpha = 16°$, $V_1 = 800\,m/s$, $K = 0.9$, $\theta = \phi$, $m = 800\,kg/h$

(i) Inlet angles of blade:

Blade velocity, $V_b = \frac{\pi DN}{60} = \frac{\pi \times 1 \times 5000}{60} = 261.80\ m/s$

From triangle ABC,

$$AC = V_{w1} = AB\cos\alpha = V_1\cos\alpha = 800 \times \cos 16 = 769.01\ m/s$$

$$BC = V_{f1} = AB\sin\alpha = V_1\sin\alpha = 800 \times \sin 16° = 220.51\ m/s$$

$$DC = AC - AD = 769.01 - 261.80 = 507.21\ m/s$$

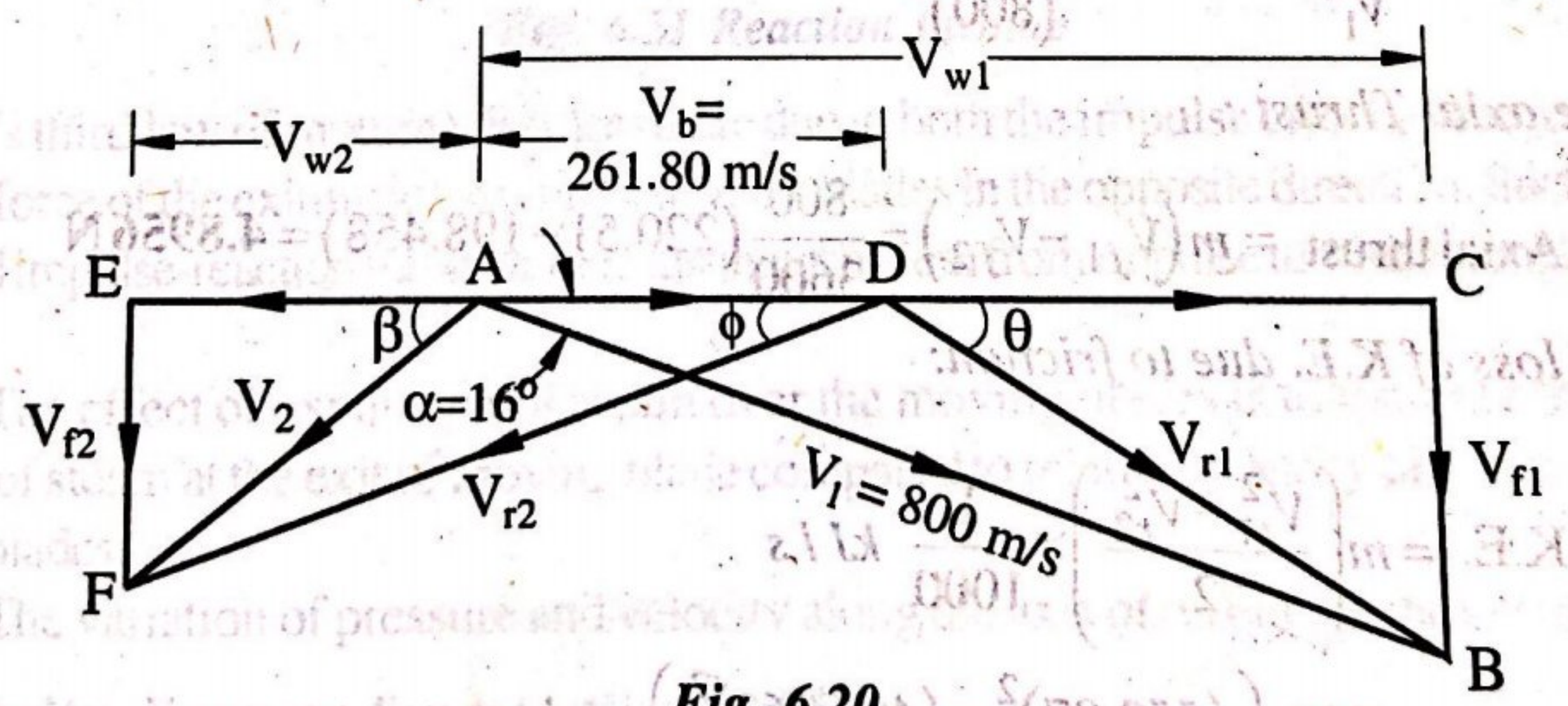


Fig. 6.20

$$\tan\theta = \frac{BC}{DC} = \frac{220.51}{507.21} = 0.43475$$

Note : Refer paper of Exam – Summer 2013 at the end of this book (Q.3(b))

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Ans

$\therefore$ Inlet angle of moving blade, $\theta = \mathbf{23.497°}$

$$V_{r1} = \sqrt{DC^2 + BC^2} = \sqrt{(507.21)^2 + (220.51)^2} = 553.07\ m/s$$

$$V_{r2} = KV_{r1} = 0.9 \times 553.07 = 497.763\ m/s$$

From triangle DFE, $DE = V_{r2}\cos\phi = 497.763 \cos 23.497 \quad (\because \theta = \phi)$

$$= 456.489\ m/s$$

$$AE = V_{w2} = DE - V_b = 456.489 - 261.80 = 194.689\ m/s$$

$$FE = V_{f2} = V_{r2}\sin\phi = 497.763 \times \sin 23.497 = 198.458\ m/s$$

$$\therefore \tan\beta = \frac{FE}{AE} = \frac{198.458}{194.689} = 1.0193$$

Inlet angle of fixed blade (outlet angle of moving blade)

$\beta = \mathbf{45.549°}$ **Ans**

(ii) Diagram Power :

$$V_w = V_{w1} + V_{w2} = 769.01 + 194.689 = 963.699\ m/s$$

$\therefore$ Diagram Power, $P = \dfrac{m\ V_w\ V_b}{1000}$

$$= \frac{800}{3600} \times \frac{963.699 \times 261.80}{1000} = \mathbf{56.066\ kW}$$ **Ans**

(iii) Diagram efficiency, η_b ;

$$\eta_b = \frac{2V_w V_b}{V_1^2} = \frac{2 \times 963.699 \times 261.80}{(800)^2} = 0.7884 = \mathbf{78.84\%}$$ **Ans**

(iv) The axial Thrust :

$$\text{Axial thrust} = m\left(V_{f1} - V_{f2}\right) = \frac{800}{3600}(220.51 - 198.458) = \mathbf{4.8956\ N}$$ **Ans**

(v) The loss of K.E. due to friction:

$$\text{Loss of K.E.} = m\left(\frac{V_{r1}^2 - V_{r2}^2}{2}\right) \times \frac{1}{1000}\ kJ/s$$

$$= \frac{800}{3600}\left(\frac{(553.07)^2 - (497.763)^2}{2}\right) \times \frac{1}{1000}$$

$= \mathbf{6.457\ kJ/s}$ **Ans**

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6.7 Reaction Turbine

In reaction turbines, pressure drop occurs both in the nozzles or the fixed row of blades, as well as in the moving row of blades, since the moving blade channels are also the nozzle shape. Due to the expansion of steam while flowing through the blades, there is an increase in kinetic energy, which gives rise to reaction in the opposite direction (according to

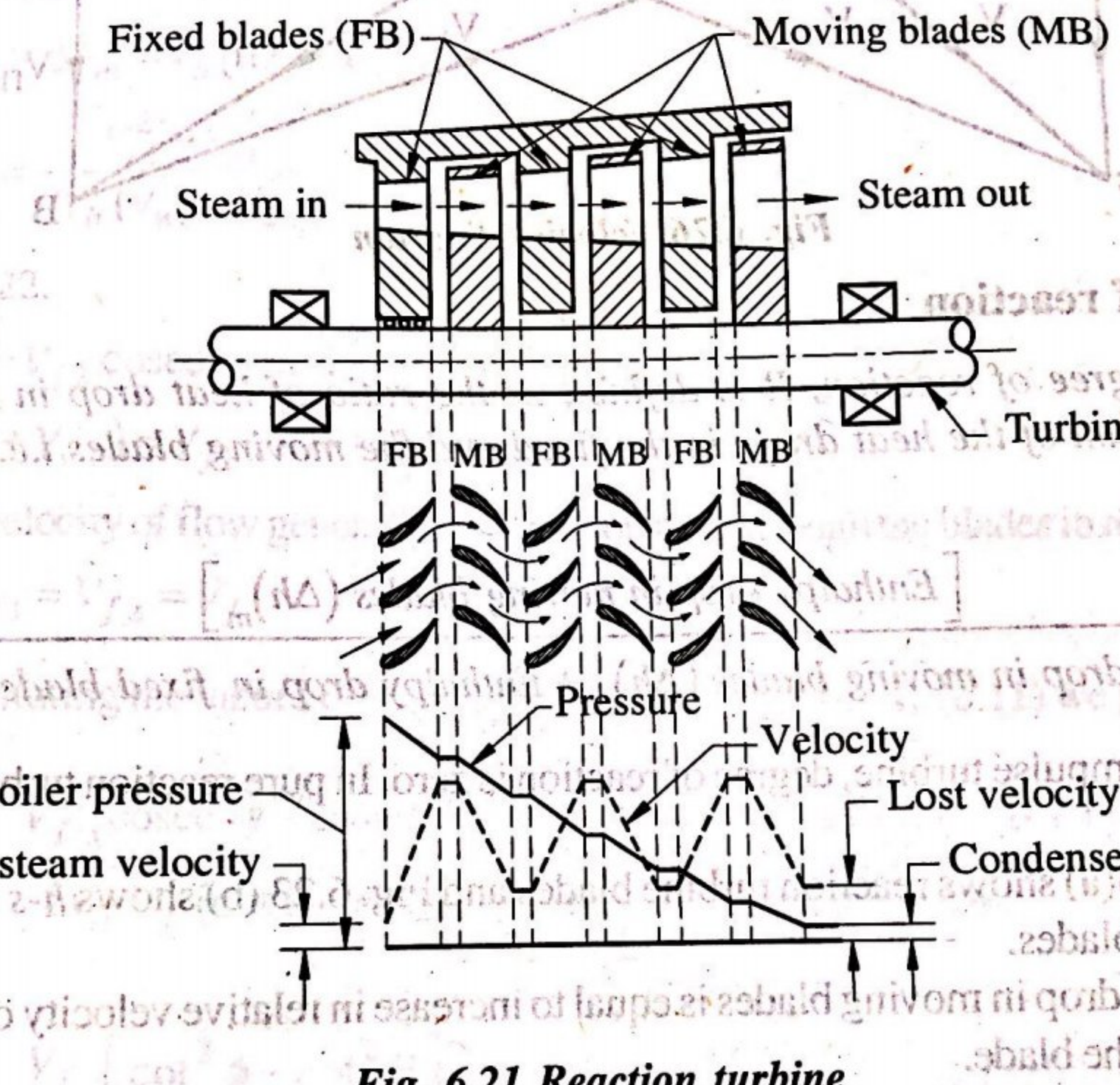


Fig. 6.21 Reaction turbine

Newton's third law of motion). Blades rotate due to both the impulse effect of the jets and the reaction force of the exiting jets impressed on the blades in the opposite direction. Such turbines are called impulse-reaction turbines, or to distinguish them from impulse turbines, simply reaction turbines.

The effect of expansion of steam over the moving blades is to increase the relative velocity of steam at the exit of moving blade compared to relative velocity of steam at entry to moving blades.

The variation of pressure and velocity along the axis of turbine is shown in Fig. 6.21

6.8 Velocity diagram for reaction turbine

The velocity diagram for reaction turbine blade is shown in Fig. 6.22. In case of an impulse turbine blade the relative velocity of steam either remains constant as the steam glides over the blades or is reduced slightly due to friction. In reaction turbine blades, the steam continuously expands as it flows over the blades. The effect of the continuous expansion of

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steam during the flow over the blade is to increase the relative velocity of steam.

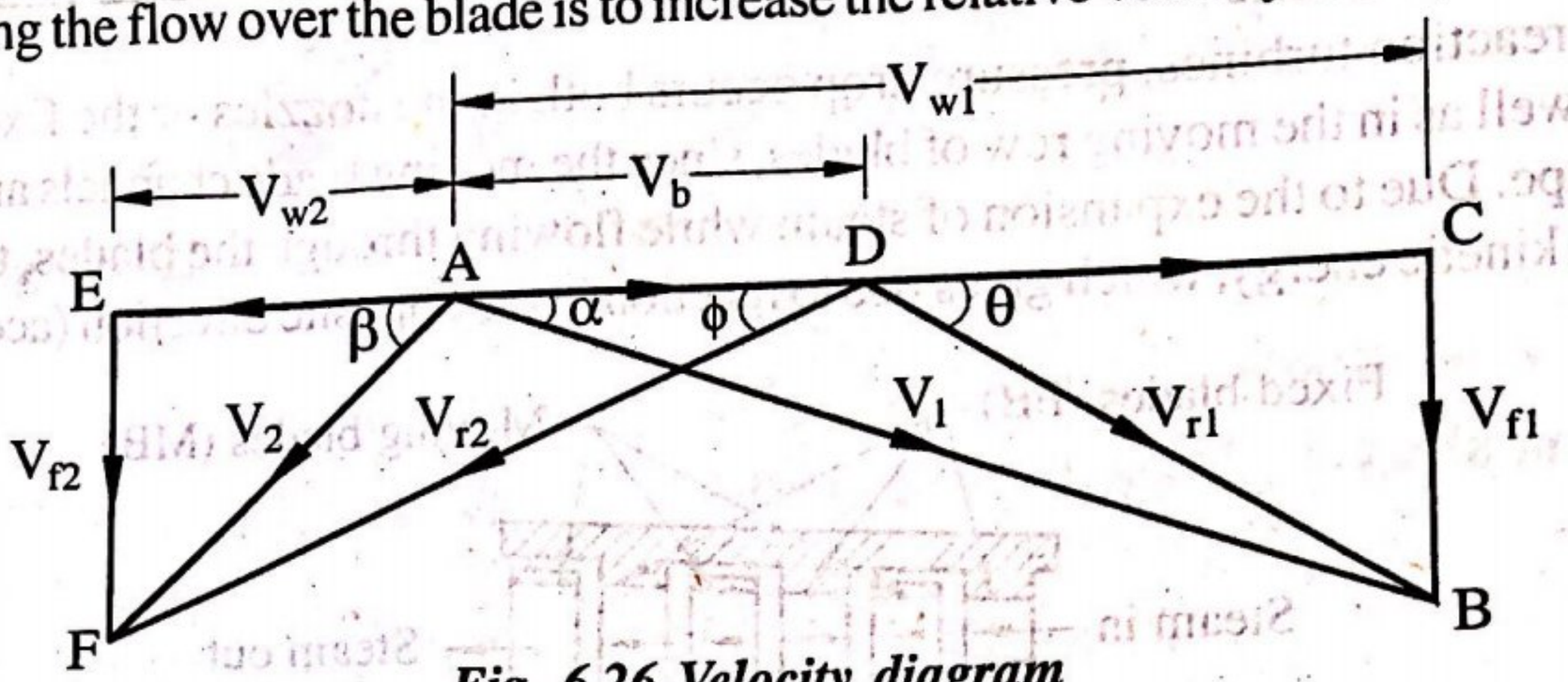


Fig. 6.26 Velocity diagram

[May '12]

6.9 Degree of reaction

The degree of reaction, R is defined as the ratio of heat drop in the moving blades to the sum of the heat drops in the fixed and the moving blades i.e. in a stage.

Mathematically,

$$R = \frac{\left[\text{Enthalpy drop in moving blades } (\Delta h)_m\right]}{\left[\text{Enthalpy drop in moving blades } (\Delta h)_m + \text{Enthalpy drop in fixed blades } (\Delta h)_f\right]}$$

In pure impulse turbine, degree of reaction is zero. In pure reaction turbine, degree of reaction = 1.

Fig. 6.23(a) shows reaction turbine blades and Fig. 6.23 (b) shows *h-s* diagram of reaction turbine blades.

The heat drop in moving blades is equal to increase in relative velocity of steam passing through the blade.

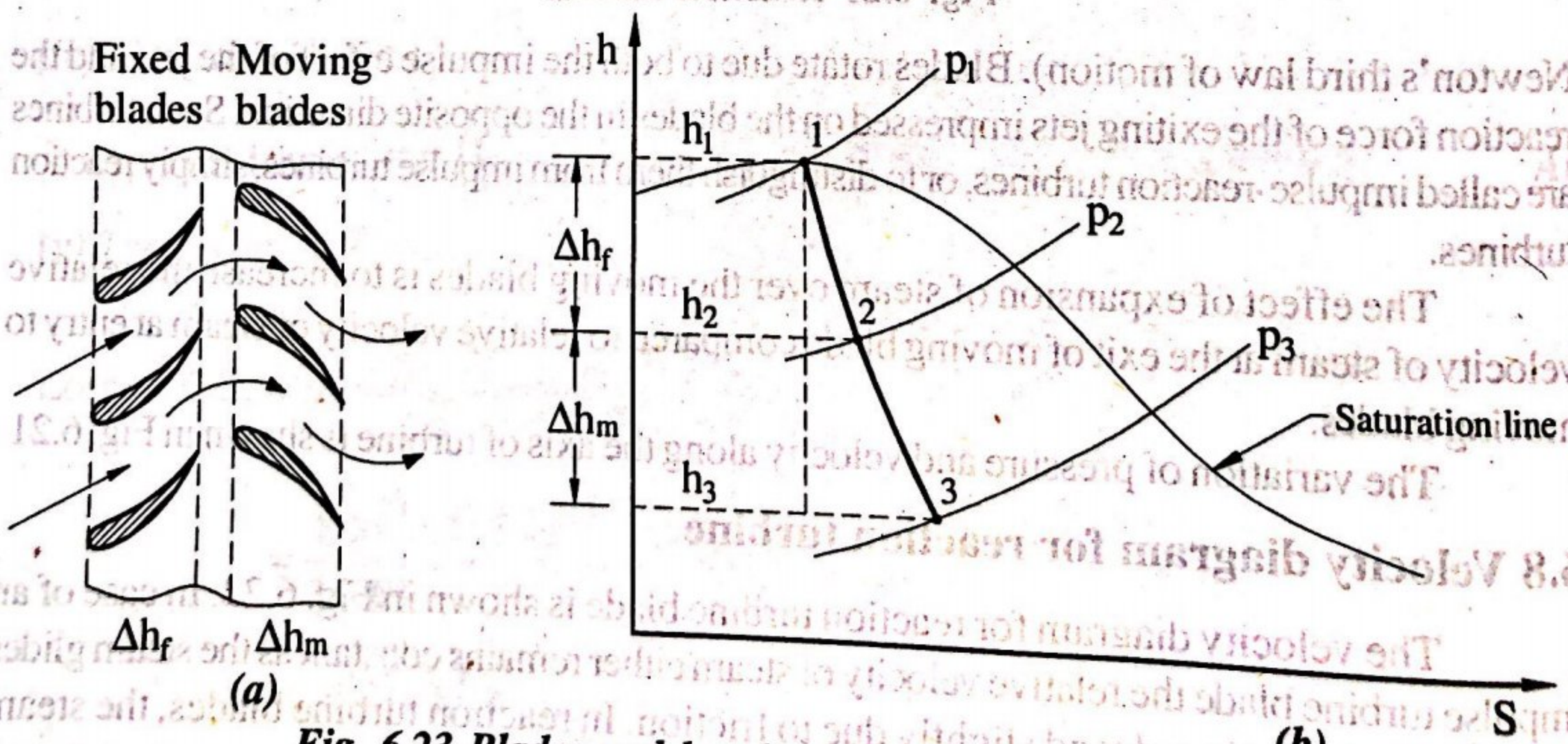


Fig. 6.23 Blades and h-s diagram of reaction turbine

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$$\therefore \Delta h_m = \frac{V_{r2}^2 - V_{r1}^2}{2}$$

The total heat drop in the stage $\left(\Delta h_f + \Delta h_m\right)$ is equal to the work done by the steam in the stage and it is given by

$$\Delta h_f + \Delta h_m = V_b \left(V_{w1} + V_{w2}\right)$$

$$\therefore R = \frac{V_{r2}^2 - V_{r1}^2}{2V_b \left(V_{w1} + V_{w2}\right)} \quad \ldots(6.11)$$

From Fig. 6.22,

$$V_{r2} = V_{f2} \operatorname{cosec} \phi \quad \text{and} \quad V_{r1} = V_{f1} \operatorname{cosec} \theta$$

$$\left(V_{w1} + V_{w2}\right) = V_{f1} \cot\theta + V_{f2} \cot\phi$$

The velocity of flow generally remains constant through the blades in reaction turbine.

$$\therefore V_{f1} = V_{f2} = V_f$$

Substituting the values of V_{r1}, V_{r2} *and* $\left(V_{w1} + V_{w2}\right)$ in eq. (6.11) we get

$$R = \frac{V_f^2 \left(\operatorname{cosec}^2\phi - \operatorname{cosec}^2\theta\right)}{2V_b\, V_f \left(\cot\theta + \cot\phi\right)} = \frac{V_f}{2V_b}\left[\frac{\left(\cot^2\phi + 1\right) - \left(\cot^2\theta + 1\right)}{\cot\theta + \cot\phi}\right]$$

$$= \frac{V_f}{2V_b}\left[\frac{\cot^2\phi - \cot^2\theta}{\cot\phi + \cot\theta}\right]$$

$$\therefore \boxed{R = \frac{V_f}{2V_b}\left(\cot\phi - \cot\theta\right)} \quad \ldots(6.12)$$

[May '12]

6.10 Parson's turbine

Parson's turbine is a particular case of reaction turbine in which the degree of reaction is half. The section of blades of this turbine is the same in both fixed and moving rows of blades.

In the Parson's turbine, the blade section and the mean diameter of fixed as well as the moving blades are the same and the blade height is progresively so increased such that the velocity of steam at exit from each row of blades is uniform throughout the stage. thus, the velocity triangle at the inlet and outlet of moving blades will be similar.

The Parson's turbine is designed for 50% reaction $\left(\Delta h_f = \Delta h_m\right)$, then the equation

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of degree of reaction can be written as

$$R = \frac{1}{2} = \frac{V_f}{2V_b}(\cot\phi - \cot\theta)$$

$$\therefore V_b = V_f(\cot\phi - \cot\theta) \quad ...(6.13)$$

Also V_b can be written as

$$V_b = V_f(\cot\phi - \cot\beta) \quad ...(6.14)$$

and $$V_b = V_f(\cot\alpha - \cot\theta) \quad (\because V_{f1} = V_{f2} = V_f) \quad ...(6.15)$$

Comparing the equations (6.13), (6.14) and (6.15)

$$\theta = \beta \text{ and } \phi = \alpha$$

This condition gives symmetrical velocity diagrams as shown in Fig. 6.24.

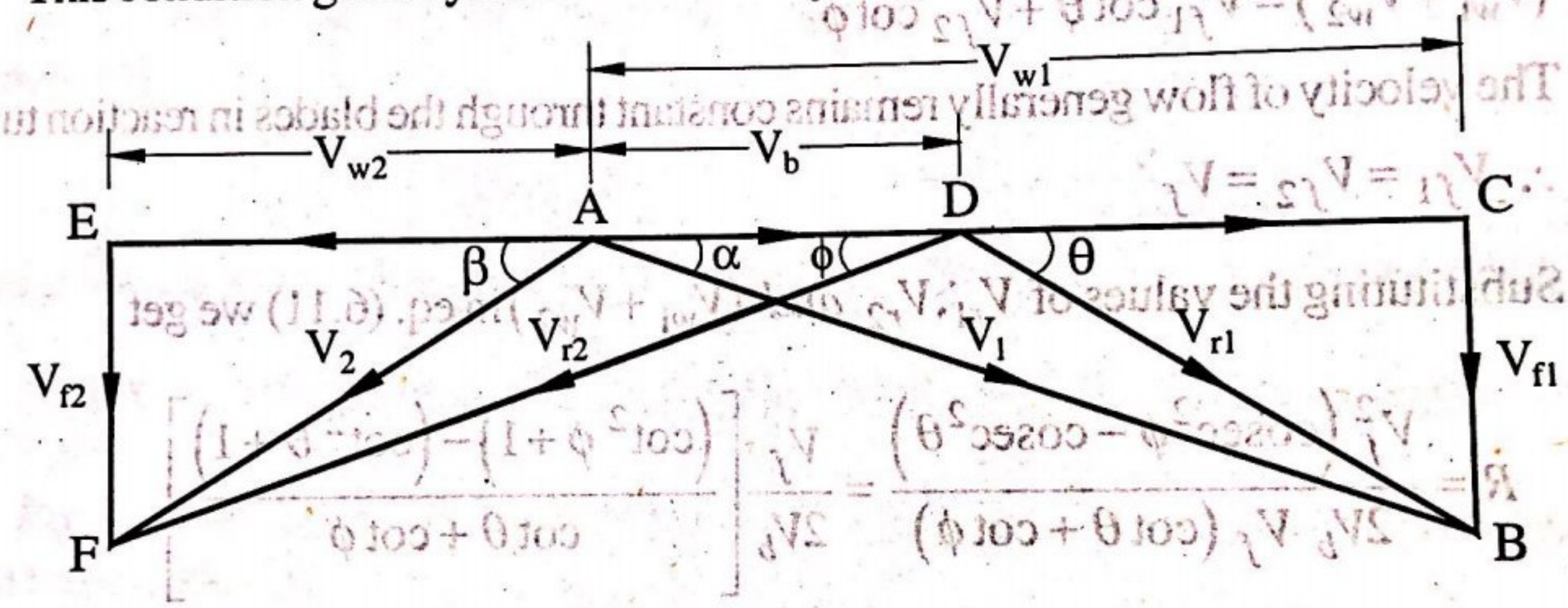


Fig. 6.24 Velocity diagrams for Parson's reaction turbine

6.11 Condition for maximum efficiency of Parson's reaction turbine [May '14]

Refer Fig. 6.24 for velocity diagrams of Parson's reaction turbine.

Work done per kg of steam is given by

$$W = V_b(V_{w1} + V_{w2}) = V_b\left[V_1\cos\alpha + (V_{r2}\cos\phi - V_b)\right]$$

For Parson's turbine, $\phi = \alpha$ and $V_{r2} = V_1$

$$\therefore W = V_b\left[2V_1\cos\alpha - V_b\right] = V_1^2\left[\frac{2V_bV_1\cos\alpha}{V_1^2} - \frac{V_b^2}{V_1^2}\right]$$

$$= V_1^2\left[2\rho\cos\alpha - \rho^2\right] \quad ...(6.16)$$

where $\rho = \frac{V_b}{V_1}$

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The kinetic energy supplied to the fixed blade $= \frac{V_1^2}{2g}$

The kinetic energy supplied to the moving blade $= \frac{V_{r2}^2 - V_{r1}^2}{2}$

$\therefore$ Total energy supplied to the stage, $\Delta h = \frac{V_1^2}{2} + \frac{V_{r2}^2 - V_{r1}^2}{2}$

For symmetrical triangles (Parson's turbine) $V_{r2} = V_1$

$$\therefore \Delta h = \frac{V_1^2}{2} + \frac{V_1^2 - V_{r1}^2}{2} = V_1^2 - \frac{V_{r1}^2}{2} \quad ...(6.17)$$

Considering the ΔABD in Fig. 6.24

$$V_{r1}^2 = V_1^2 + V_b^2 - 2V_1 V_b \cos\alpha$$

$$\therefore \Delta h = V_1^2 - \left(V_1^2 + V_b^2 - 2V_1 V_b \cos\alpha\right)/2$$

$$= \frac{V_1^2 + 2V_1 V_b \cos\alpha - V_b^2}{2} = \frac{V_1^2}{2}\left[1 + \frac{2V_b}{V_1}\cos\alpha - \left(\frac{V_b}{V_1}\right)^2\right]$$

$$\therefore \Delta h = \frac{V_1^2}{2}\left[1 + 2\rho\cos\alpha - \rho^2\right] \quad ...(6.18)$$

The blade efficiency of the reaction turbine is given by

$$\eta_b = \frac{Work\ done\ (W)}{Total\ energy\ supplied\ (\Delta h)}$$

$$= \frac{V_1^2\left(2\rho\cos\alpha - \rho^2\right)}{\frac{V_1^2}{2}\left(1 + 2\rho\cos\alpha - \rho^2\right)}$$

$$= \frac{2\left(2\rho\cos\alpha - \rho^2\right)}{\left(1 + 2\rho\cos\alpha - \rho^2\right)} = \frac{2\rho\left(2\cos\alpha - \rho\right)}{\left(1 + 2\rho\cos\alpha - \rho^2\right)} = \frac{2\left(1 + 2\rho\cos\alpha - \rho^2\right) - 2}{\left(1 + 2\rho\cos\alpha - \rho^2\right)}$$

$$= 2 - \frac{2}{1 + 2\rho\cos\alpha - \rho^2} \quad ...(6.19)$$

The η_b becomes maximum when factor $\left(1 + 2\rho\cos\alpha - \rho^2\right)$ becomes maximum,

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$\therefore$ The required condition is

$$\frac{d}{d\rho}\left(1+2\rho\cos\alpha-\rho^2\right)=0$$

$$\therefore 2\cos\alpha-2\rho=0$$

$$\therefore \rho=\cos\alpha \qquad ...(6.20)$$

i.e. The condition for maximum efficiency, $\rho=\frac{V_b}{V_1}=\cos\alpha$.

Substituting the value of ρ in equation (6.19), we get

$$(\eta_b)_{max}=2-\frac{2}{1+2\cos\alpha\cos\alpha-\cos^2\alpha}$$

$\therefore$ Maximum efficiency, $(\eta_b)_{max}=\frac{2\cos^2\alpha}{1+\cos^2\alpha}$ ...(6.21)

(1) Stage efficiency :

Stage efficiency is given by

$$\eta_{stage}=\frac{\text{Net work done on shaft per stage per kg of steam flowing}}{\text{Adiabatic heat drop per stage}}$$

$$\eta_{stage}=\frac{(V_{w1}+V_{w2})V_b}{(\Delta h)\times 1000}$$

where, Δh is the heat drop in kJ/kg in a stage which can be determined with the help of Mollier's diagram. Also

$$\Delta h=\frac{V_1^2}{2}+\frac{V_{r2}^2-V_{r1}^2}{2}$$

(2) Blade or diagram efficiency :

It is the ratio of work done on the blade per second to the energy entering the blade per second.

$$\therefore \eta_b=\frac{\text{Work done}}{\text{Energy supplied}}=\frac{(V_{w1}+V_{w2})V_b}{\left(\frac{V_1^2}{2}+\frac{V_{r2}^2-V_{r1}^2}{2}\right)}$$

$$\therefore \eta_b=\frac{2(V_{w1}+V_{w2})V_b}{V_1^2+\left(V_{r2}^2-V_{r1}^2\right)}$$

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(3) Blade height :

In the reaction turbine, the steam enters the moving blades over the whole circumference. As a result of this, the area through which the steam flows is always full of steam. Now consider a reaction turbine whose end view of the blade ring is shown in Fig. 6.25. Let,

N = Speed of turbine, *rpm*

D = Drum diameter, *m*

h = Blade height, *m*

$V_f = V_{f1} = V_{f2}$ = Velocity of flow, m/s.

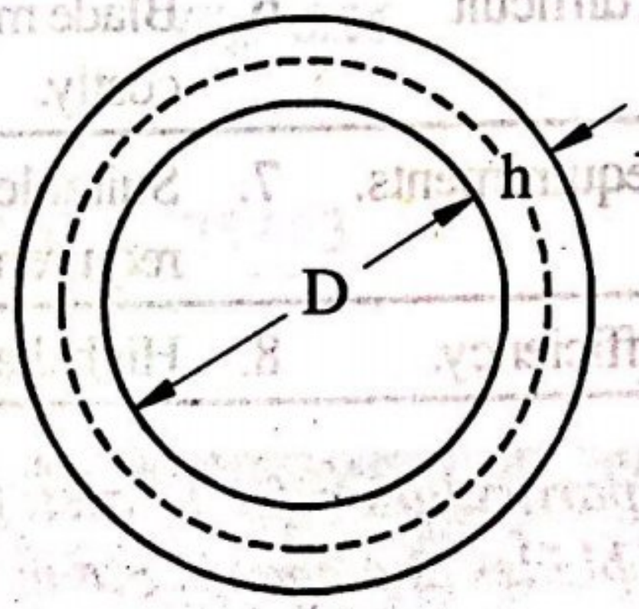


Fig. 6.25

∴ Mean drum diameter, $D_m = (D + h)$

∴ Total area available for the steam to flow, $A_f = \pi (D + h) h$

Volume flow rate of steam = Area of flow x velocity of flow

$$= [\pi (D + h) h] \times V_f$$

Mass flow rate of steam,

$$m = \frac{[\pi (D + h) h] \times V_f}{v} \quad ...(6.22)$$

where 'v' is specific volume of steam in m^3/kg at the given stage pressure.

Blade speed is given by

$$V_b = \frac{\pi (D + h) N}{60} m/s$$

• Comparison between impulse and reaction turbines [Nov.'14, May '13]

	Impulse Turbine		Reaction Turbine
1.	Steam expands in nozzles only, no expansion in moving blades.	1.	Steam expands in both fixed and moving blades.
2.	Blade channels area is constant.	2.	Blade channels area is converging types.

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3. Profile types symmetrical blades are used.	3. Aerofoil type blades are used.
4. Occupies less space for same power output.	4. Occupies more space for same power output.
5. If friction is neglected relative velocity of steam at inlet and outlet of moving blades is same.	5. Relative velocity at outlet of moving blades is always greater than that at the inlet.
6. Blade manufacturing is not difficult and thus not costly	6. Blade manufacturing is difficult and hence costly.
7. Suitable for small power requirements.	7. Suitable for medium and higher power requirements.
8. Relatively lower diagram efficiency.	8. High diagram efficiency.

Problem 6.9 *The following particulars refer to a stage of a Parson's steam turbine comprising one ring of fixed blades and one ring of moving blades; Mean diameter of blade ring = 80 cm, turbine speed = 3000 rpm, steam velocity at exit from blades = 180 m/s, Blade outlet angle = 16°, steam flow through blades = 10 kg/s. Draw the velocity diagram and find the following : (1) blade inlet angle, (ii) tangential force on the ring of moving blades, (iii) power developed in a stage.* **[Same as May '12]**

Solution **:** Given data

$D_m = 80\ cm = 0.8\ m,\ N = 3000\ rpm,\ \alpha = 16° = \phi,\ V_1 = 180\ m/s,\ m = 10\ kg/s,\ \theta = ?,$

$F = ?,\ P = ?$

$$\text{Blade velocity, } V_b = \frac{\pi D_m N}{60} = \frac{\pi \times 0.8 \times 3000}{60}$$

$$= 125.663 \text{ m/s}$$

(i) Blade inlet angle :

The inlet velocity triangle ABD can be drawn with the values of V_b, α and V_1 as shown in Fig. 6.26.

This gives, $\theta = 47°$ *Ans*

In Parson's reaction turbine, $V_{r2} = V_1, V_2 = V_{r1}, \theta = \beta, \phi = \alpha$, so the outlet velocity triangle can be drawn.

(ii) Tangential force :

From velocity diagram, $CE = V_w = 220.09\ m/s$

$$F = m \cdot V_w = 10 \times 220.09 = 2200.9 \text{ N}$$ *Ans*

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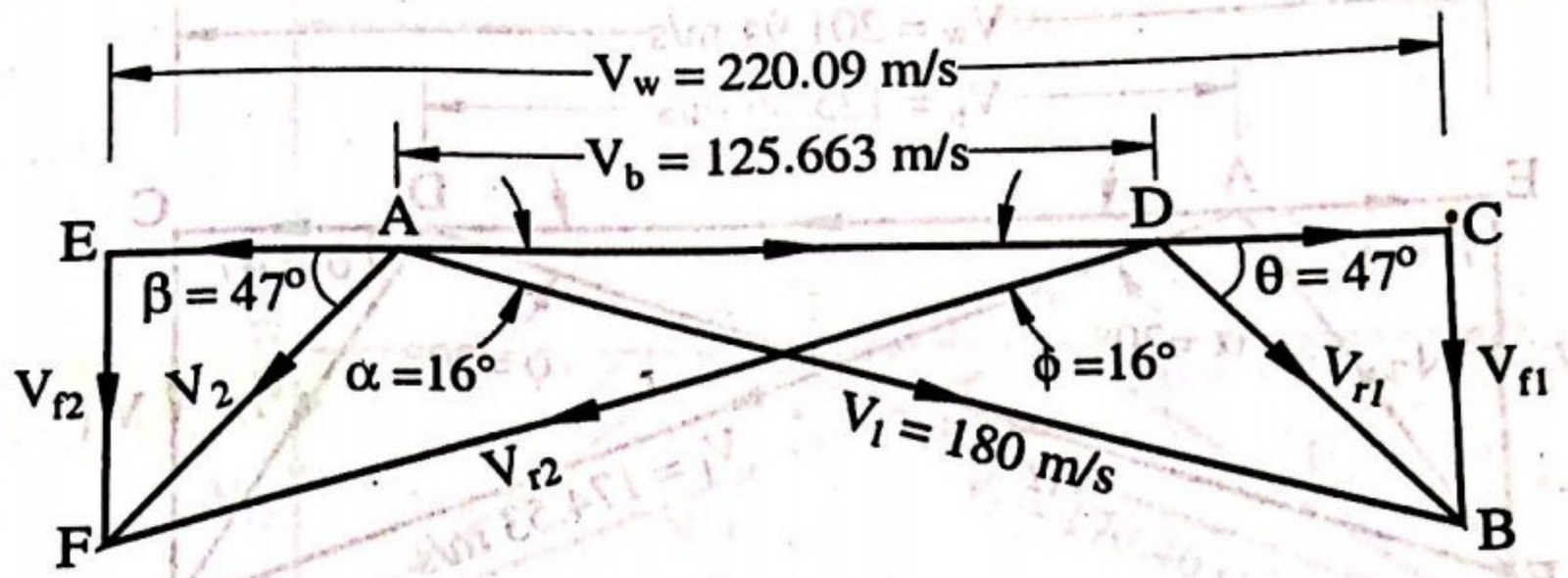


Fig. 6.26

(iii) Power developed :

$$P = \frac{m\ V_w\ V_b}{1000} = \frac{10 \times 220.09 \times 125.663}{1000}$$

$$= 276.57 \text{ kW}$$

Ans

This problem can be solved by analytical method also.

Problem 6.10 *The following data refer to a particular stage of a Parson's reaction turbine. Speed of the turbine = 2000 rpm, Mean diameter of the rotor = 1.2 m, stage efficiency = 83%, Blade outlet angle = 20°, speed ratio = 0.72. Determine the available isentropic enthalpy drop in the stage.*

Solution : Given data

$D_m = 1.2\ m,\ N = 2000\ rpm,\ \eta_{stage} = 83\%,\ \phi = 20°,\ \rho = \frac{V_b}{V_1} = 0.72,\ \Delta h = ?$

Blade speed, $V_b = \frac{\pi D_m N}{60} = \frac{\pi \times 1.2 \times 2000}{60} = 125.66\ m/s$

$$\rho = \frac{V_b}{V_1} = 0.72 = \frac{125.66}{V_1}$$

$$V_1 = 174.53\ m/s$$

For Parson's Turbine, $\alpha = \phi$

With the above data known, the velocity diagram for the turbine can be drawn as shown in Fig. 6.27.

Measuring from the diagram,

$$V_w = 201.94\ m/s$$

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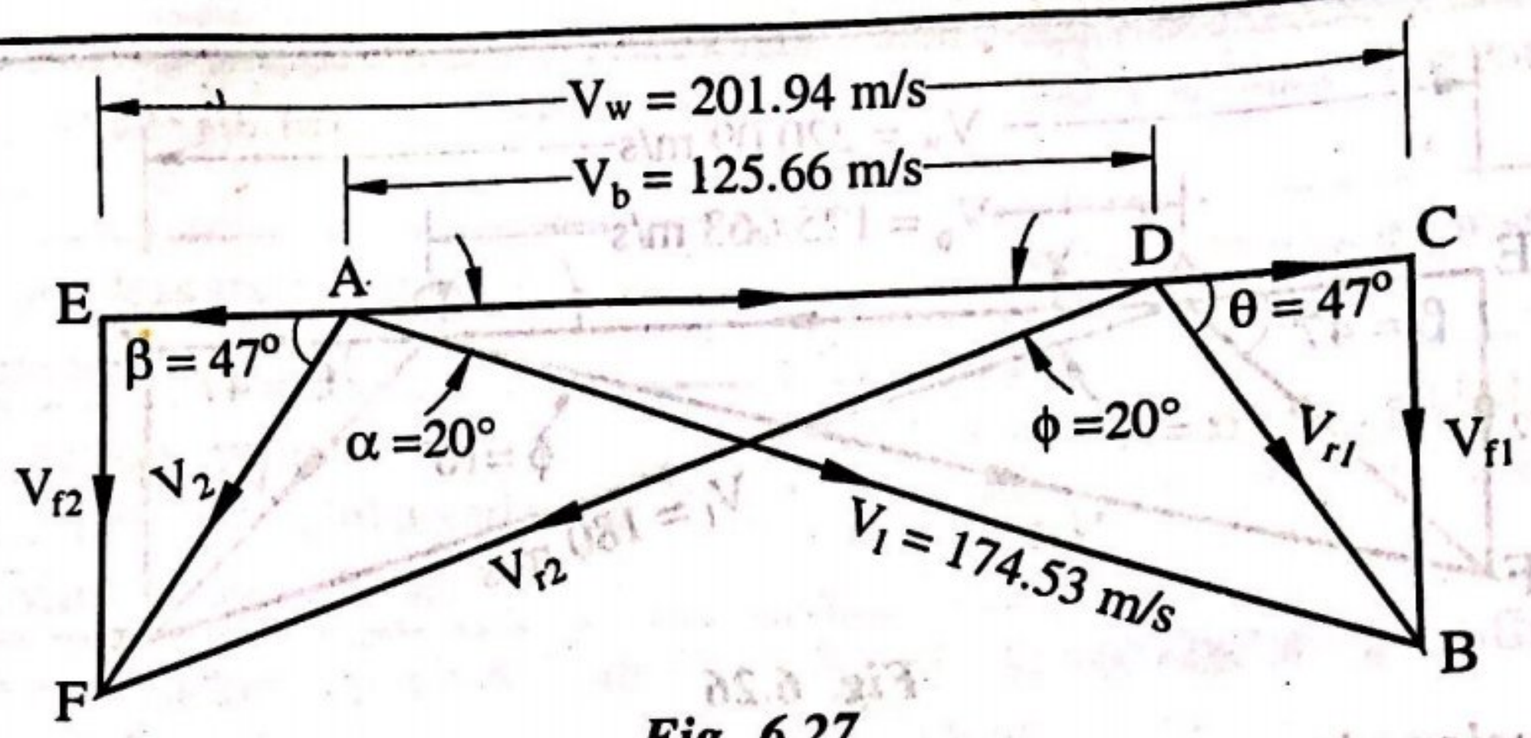


Fig. 6.27

Stage efficiency is given by

$$\eta_{stage} = \frac{V_w\, V_b}{\Delta h \times 1000}$$

$$\therefore 0.83 = \frac{201.94 \times 125.66}{\Delta h \times 1000}$$

$$\therefore \Delta h = \mathbf{30.573\,kJ}$$ **Ans**

Problem 6.11 *The angles at inlet and discharge of the blading of a 50% reaction turbine are 36° and 22° respectively. The speed of rotation is 3000 rpm and at a particular stage, the mean ring diameter is 0.45 m and the steam condition is at 1.7 bar pressure and 0.95 dry. Determine (i) the required height of blading to pass 4 kg/s of steam, and (ii) the power developed by the ring.*

Solution : Given data

$\alpha = \phi = 22°,\ \theta = \beta = 36°,\ N = 3000\,rpm,\ D_m = 0.45\,m,\ p_1 = 1.7\,bar,\ x_1 = 0.95,$

$m = 4\ kg/s,\ h = ?,\ p = ?$

$$\textit{Blade speed, } V_b = \frac{\pi D_m N}{60} = \frac{\pi \times 0.45 \times 3000}{60} = 70.68\ m/s$$

Graphical method :

The velocity diagram can be drawn with the above known data. The velocity diagram is shown in Fig. 6.28.

Measuring from diagram,

$$V_1 = 170.46\ m/s = V_{r2},\ V_{r1} = 108.23\ m/s,\ V_w = 245.47\ m/s,$$

$$V_{f1} = V_{f2} = V_f = 63.92\ m/s$$

From steam tables at $p_1 = 1.7$ bar pressure.

$$v_g = 1.031\ m^3/kg$$

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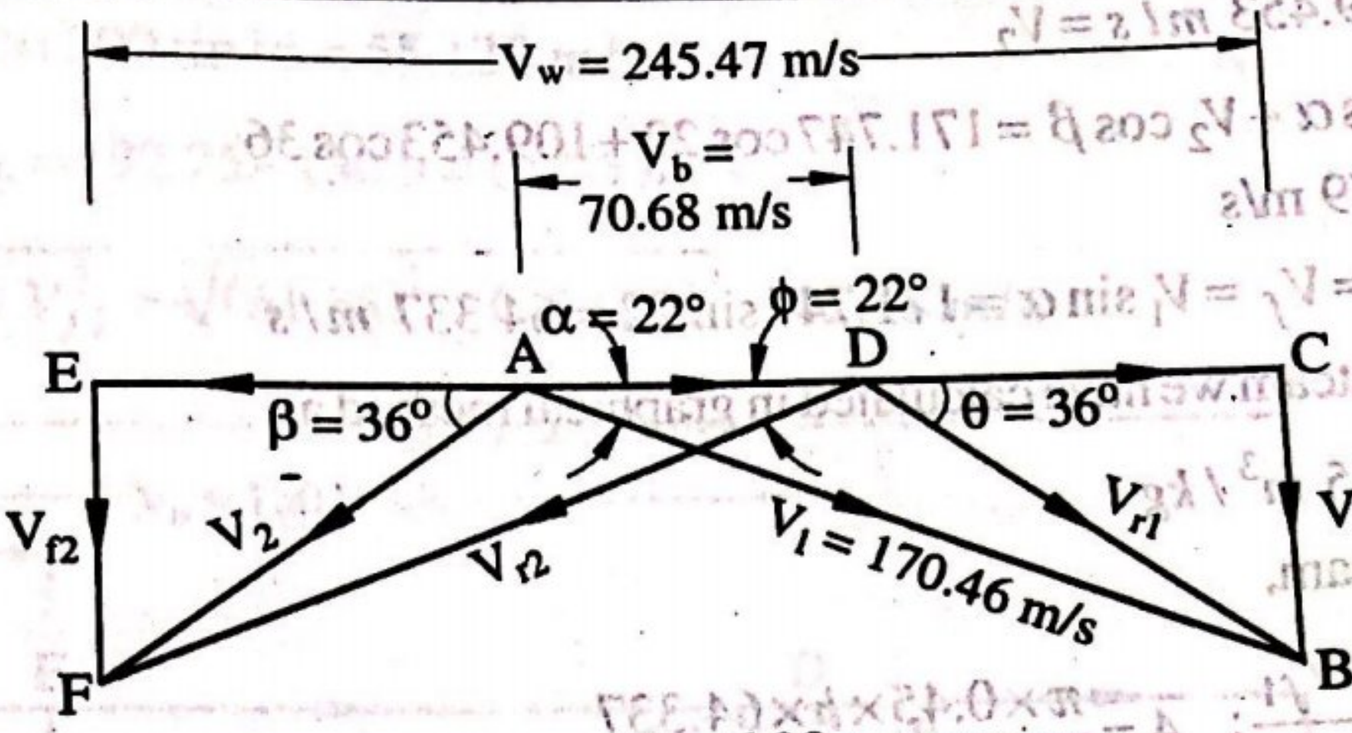


Fig. 6.28

$\therefore$ Specific volume of steam, $v_1 = xv_g = 0.95 \times 1.031 = 0.97945\ m^3/kg$

Mass flow rate of steam,

$$m = \frac{\pi D_m h V_f}{v_1}$$

$$\therefore 4 = \frac{\pi \times 0.45 \times h \times 63.92}{0.97945}$$

$$\therefore h = 0.04335m = \mathbf{43.35\,mm}$$ **Ans**

Power developed by the ring,

$$P = \frac{m V_w V_b}{1000} = \frac{4 \times 245.47 \times 70.68}{1000} = \mathbf{69.4\ kW}$$ **Ans**

Analytical method :

By sine law,

$$\frac{V_1}{\sin(180-\theta)} = \frac{V_b}{\sin 14} = \frac{V_{r1}}{\sin \alpha}$$

$$\therefore \frac{V_1}{\sin 144} = \frac{V_b}{\sin 14} = \frac{V_{r1}}{\sin 22}$$

$$\therefore V_1 = \frac{V_b \times \sin 144}{\sin 14} = \frac{70.68 \times 0.5878}{0.2419}$$

$$\therefore V_1 = 171.747\ m/s = V_{r2}, \text{ and}$$

$$V_{r1} = \frac{V_b \times \sin 22}{\sin 14} = \frac{70.68 \times 0.3746}{0.2419}$$

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$$\therefore V_{r1} = 109.453\ m/s = V_2$$

$$V_w = V_1 \cos\alpha + V_2 \cos\beta = 171.747 \cos 22 + 109.453 \cos 36$$
$$= 247.79\ \text{m/s}$$

$$V_{f1} = V_{f2} = V_f = V_1 \sin\alpha = 171.747 \sin 22 = 64.337\ m/s$$

Specific volume of steam we have calculated in graphical method as

$$v_1 = 0.97945\ m^3/kg$$

Mass flow rate of steam,

$$m = \frac{\pi D_m\ h\ V_{f1}}{v_1};\quad 4 = \frac{\pi \times 0.45 \times h \times 64.337}{0.97945}$$

$$\therefore\ h = 0.043074\ m = \mathbf{43.074\ mm} \qquad \textit{Ans}$$

Power developed by the ring,

$$p = \frac{m\ V_w\ V_b}{1000} = \frac{4 \times 247.79 \times 70.68}{1000}$$
$$= \mathbf{70.055\ kW} \qquad \textit{Ans}$$

Problem 6.12 *The following particulars refer to a stage of an impulse-reaction turbine: Turbine speed = 2500 rpm, mean diameter = 100 cm, outlet angle of fixed blades = 16°, height of blades = 20 cm, specific volume of steam at exit of fixed blades = 3.8 m^3/kg and that at exit of moving blades = 4.2 m^3/kg, steam velocity at exit of fixed blades = 200 m/s, power developed = 200 kW. Calculate the enthalpy drop in the stage, degree of raction, moving blade outlet angle and gross stage efficiency. Assume carry over co-efficient as 0.85, expansion efficiency = 0.95. Thickness of blades may be neglected.*

Solution : Given data

$$N = 2500\ rpm,\ D_m = 100\ cm = 1\ m,\ \alpha = 16°,\ h = 20\ cm = 0.2\ m,\ v_i = 3.8\ m^3/kg,$$

$$v_e = 4.2\ m^3/kg,\ V_1 = 200\ m/s,\ P = 200\ kW,\ \text{carryover coefficient, } \delta = 0.85,\ \eta_{exp} = 0.95,$$

$$\Delta H = ?, R = ?, \phi = ?, \eta_{stage} = ?.$$

$$\text{Blade speed, } V_b = \frac{\pi\ D_m\ N}{60} = \frac{\pi \times 1 \times 2500}{60} = 130.9\ m/s$$

Fig. 6.29 shows the velocity diagram for the given reaction turbine.

From triangle *ABC* :

$$V_{w1} = V_1 \cos\alpha = 200 \cos 16 = 192.25\ m/s$$

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$$V_{f1} = V_1 \sin\alpha = 200 \sin 16 = 55.127 \ m/s$$

$$DC = V_{w1} - V_b = 192.25 - 130.9 = 61.35 \ m/s$$

$$V_{r1} = \sqrt{DC^2 + V_{f1}^2} = \sqrt{(61.35)^2 + (55.127)^2} = 82.48 \ m/s$$

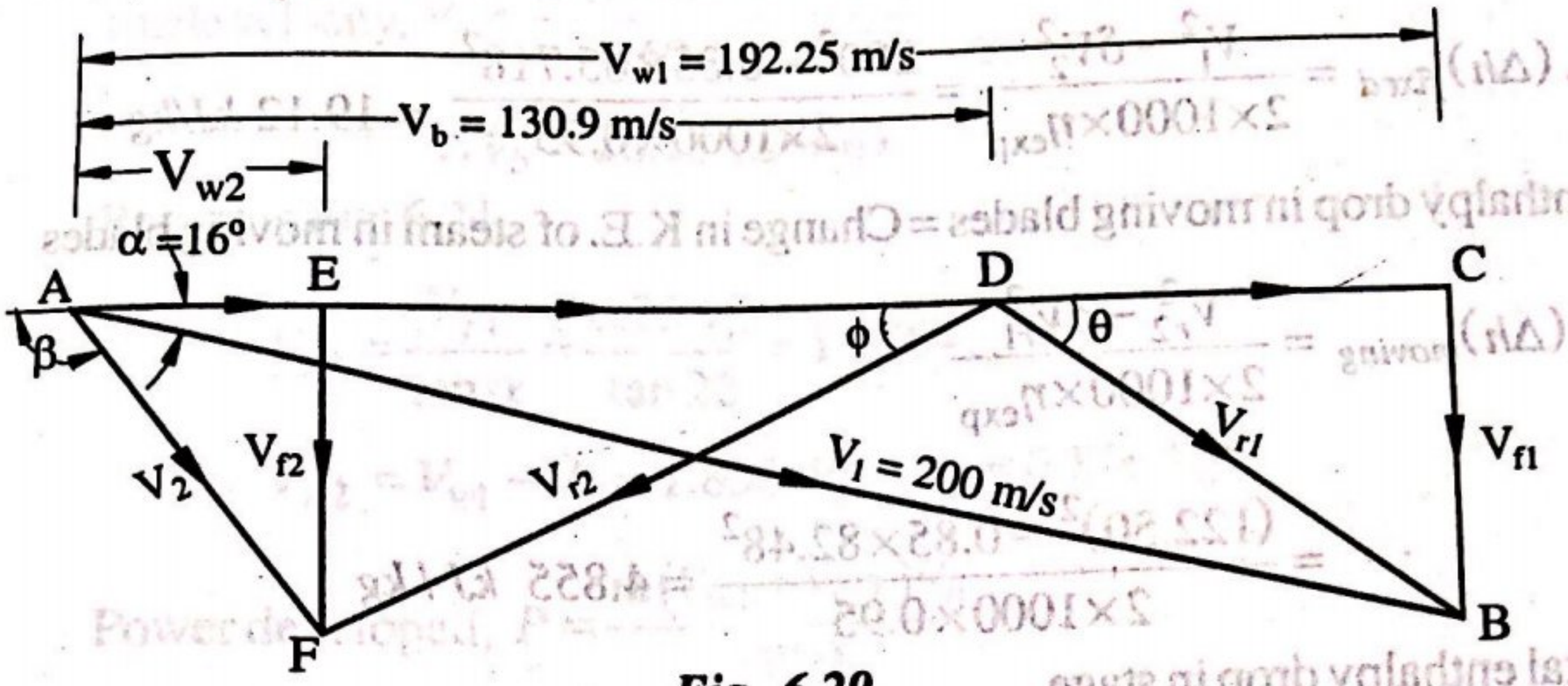


Fig. 6.29

Mass flow rate, $m = \dfrac{\pi D_m h V_{f1}}{v_i} = \dfrac{\pi \times 1 \times 0.2 \times 55.127}{3.8}$

$$\therefore m = 9.115 \ kg/s$$

Power developed, $P = \dfrac{m(V_{w1} + V_{w2})V_b}{1000} \ kW$

$$\therefore 200 = \frac{9.115(192.25 + V_{w2}) \times 130.9}{1000}$$

$$\therefore V_{w2} = -24.626 \ m/s$$

Also, $m = \dfrac{\pi D_m \ h \ V_{f2}}{v_e}$

$$9.115 = \frac{\pi \times 1 \times 0.2 \times V_{f2}}{4.2}$$

$$\therefore V_{f2} = 60.93 \ m/s$$

Therefore, the exit velocity diagram ADF can be completed as shown in Fig. 6.29

From ΔEDF, $V_{r2} = \sqrt{ED^2 + EF^2} = \sqrt{(V_b - V_{w2})^2 + V_{f2}^2}$

$$= \sqrt{(130.9 - 24.626)^2 + (60.93)^2} = 122.50 \text{ m/s}$$

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From ΔAFE, $V_2 = \sqrt{AE^2 + EF^2} = \sqrt{V_{w2}^2 + V_{f2}^2}$

$$= \sqrt{(24.626)^2 + (60.93)^2} = 65.718 \ m/s$$

Enthalpy drop in fixed row of blades = (change in K.E. of steam)

$$\therefore (\Delta h)_{fixed} = \frac{V_1^2 - \delta V_2^2}{2 \times 1000 \times \eta_{exp}} = \frac{200^2 - 0.85 \times 65.718^2}{2 \times 1000 \times 0.95} = 19.12 \ kJ/kg$$

Enthalpy drop in moving blades = Change in K.E. of steam in moving blades

$$\therefore (\Delta h)_{moving} = \frac{V_{r2}^2 - \delta V_{r1}^2}{2 \times 1000 \times \eta_{exp}}$$

$$= \frac{(122.50)^2 - 0.85 \times 82.48^2}{2 \times 1000 \times 0.95} = 4.855 \ kJ/kg$$

Total enthalpy drop in stage,

$$\Delta H = m\left[(\Delta h)_{fixed} + (\Delta h)_{moving}\right]$$

$$= 9.115[19.12 + 4.855] = \mathbf{218.53\ kJ} \quad \textit{Ans}$$

Degree of reaction, $R = \dfrac{(\Delta h)_{moving}}{(\Delta h)_{fixed} + (\Delta h)_{moving}} = \dfrac{4.855}{19.12 + 4.855}$

$$R = \mathbf{0.2025} \quad \textit{Ans}$$

Moving blade outlet angle, ϕ:

$$\sin\phi = \frac{V_{f2}}{V_{r2}} = \frac{60.93}{122.50} = 0.49738$$

$$\therefore \phi = \mathbf{29.82°} \quad \textit{Ans}$$

Gross stage efficiency,

$$\eta_{stage} = \frac{P}{\Delta H} = \frac{200}{218.53} = 0.9152 = \mathbf{91.52\%} \quad \textit{Ans}$$

Problem 6.13 *In a stage of 50% reaction turbine, the steam consumption is 6 kg/s and it runs at 500 rpm. The discharge blade tip angles are 22° both for fixed and moving blades. The axial velocity of flow is 0.75 times the blade velocity. Determine the drum diameter and blade height of a particular turbine pair where pressure of steam is 1.7 bar of dryness fraction 0.96. The power developed by the turbine amounts to 4 kW.*

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Solution : Given data :

$m = 6\ kg/s,\ N = 500\ rpm,\ \alpha = \phi = 22°, V_f = V_{f1} = V_{f2} = 0.75\ V_b,\ p = 1.7\ bar,$

$x = 0.96,\ P = 4\ kW,\ D = ?,\ h = ?$

$$\text{Blade velocity, } V_b = \frac{\pi (D+h) N}{60} = \frac{\pi (D+h) 500}{60}$$

$$\therefore V_b = 26.18\ (D+h) \quad \text{...(i)}$$

Referring Fig. 6.24,

$$V_{w1} = \frac{V_{f1}}{\tan \alpha} = \frac{0.75\ V_b}{\tan 22} = 1.856 V_b$$

$$V_{w2} = V_{w1} - V_b = 1.856\ V_b - V_b = 0.856\ V_b$$

$$\text{Power developed, } P = \frac{m (V_{w1} + V_{w2}) V_b}{1000}$$

$$\therefore 4 = \frac{6 (1.856\ V_b + 0.856\ V_b) V_b}{1000}$$

$$V_b = 15.678\ m/s$$

$$V_{f1} = 0.75\ V_b = 0.75 \times 15.678 = 11.76\ m/s$$

From equation (i)

$$15.678 = 26.18\ (D + h)$$

$$\therefore (D + h) = 0.599 \text{ m}$$

From steam tables at 1.7 bar pressure, $v_g = 1.031\ m^3/kg$

$$\text{Specific volume of steam, } v = x v_g = 0.96 \times 1.031$$

$$= 0.98976\ m^3/kg$$

$$\text{Mass flow rate, } m = \frac{\pi\ (D+h)\ h\ V_f}{v}$$

$$\therefore 6 = \frac{\pi \times 0.599 \times\ h \times 11.76}{0.98976}$$

$$\therefore h = 0.2683\ m = \mathbf{26.83\ cm}$$

Since $D + h = 0.599$

$$\therefore D + 0.2683 = 0.599$$

$$\therefore D = 0.3307\ m = \mathbf{33.07\ cm}$$

Ans

An

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Problem 6.14 *A Parsons reaction turbine develops 6 MW at speed of 500 rpm using 7 kg/kWh of steam. The exit angle of the blades are 20° and the velocity of steam is 1.30 times the blade velocity and pressure at exit is 1.3 bar and dryness fraction is 0.92. Calculate : (i) a suitable blade height, assuming the ratio of drum diameter to blade height is 8, and (ii) diagram power.*

Solution : Given data

$P = 6\,MW, N = 500\ rpm,\ m = 7\ kg/kWh, \alpha = \phi = 20°,\ V_1 = 1.30\ V_b,\ P = 1.3$ bar, $x = 0.92, D/h = 8, h = ?$, Diagram power = ?

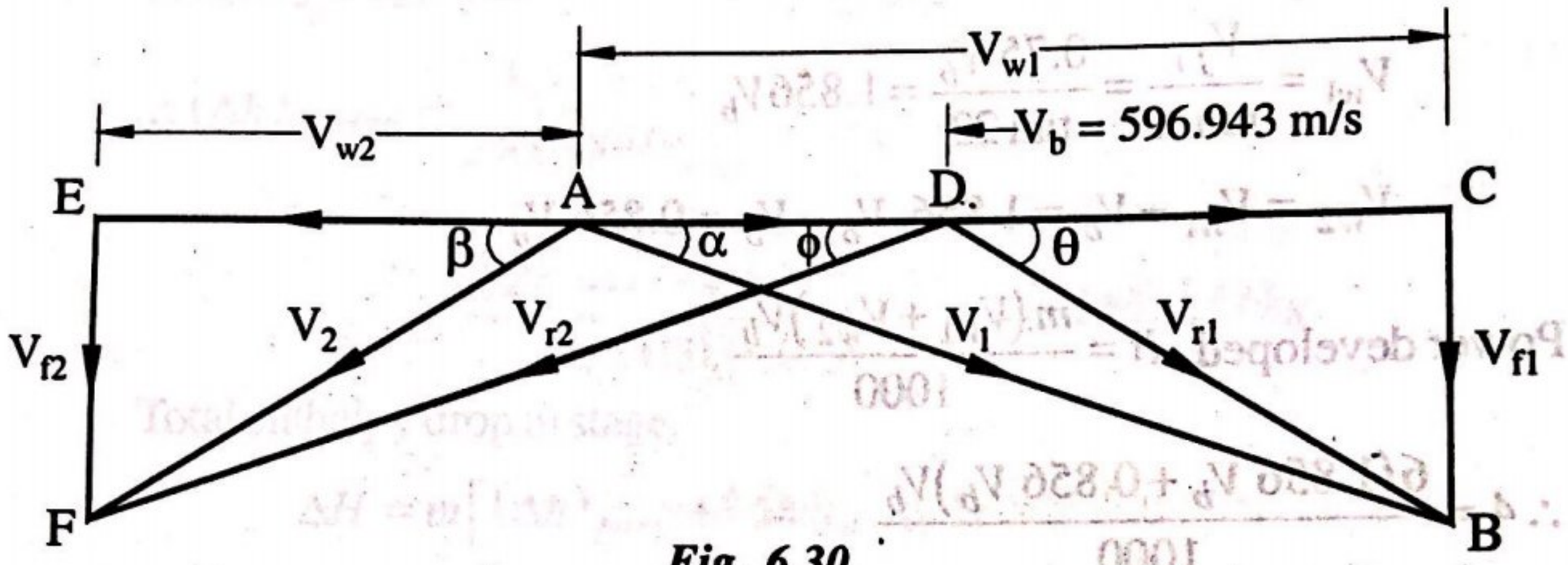


Fig. 6.30

Mass flow rate, m = steam consumption × power

$$= 7 \times 6 \times 10^3 = 42000\ kg/h = \frac{42000}{3600} = 11.667\ kg/s$$

Fig. 6.30 shows the velocity diagram.

$$V_{w1} = V_1 \cos\alpha = 1.3 V_b \cos 20 = 1.2216\ V_b$$

$$V_{w2} = V_{w1} - V_b = 1.2216 V_b - V_b = 0.2216\ V_b$$

(1) Blade height :

Power developed, $P = \dfrac{m(V_{w1} + V_{w2})V_b}{1000}$

$$6 \times 10^3 = \frac{11.667(1.2216 V_b + 0.2216 V_b)V_b}{1000}$$

$$V_b = 596.943\ m/s$$

$$\therefore V_1 = 1.30\ V_b = 1.3 \times 596.943 = 776.026\ m/s$$

$$V_f = V_{f1} = V_{f2} = V_1 \sin\alpha = 776.026 \sin 20 = 265.416\ m/s$$

From steam tables at 1.3 bar, $v_g = 1.325\ m^3/kg$

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$\therefore$ Specific volume of steam at exit,

$$v = x v_g = 0.92 \times 1.325 = 1.219 \, m^3 / kg$$

Mass flow rate, $m = \dfrac{\pi (D+h) h \, V_f}{v}$

$$\therefore 11.667 = \frac{\pi (8h+h) h \times 265.416}{1.219}$$

$$\therefore h = 0.04353 \; m = \mathbf{43.53} \; mm$$ **Ans**

(ii) Diagram power :

Diagram power represent the power input to turbine.

$$DC = AE = V_{w2} = 0.2216 \; V_b = 0.2216 \times 596.943 = 132.28 \text{ m/s}$$

$$V_{r1} = \sqrt{DC^2 + V_{f1}^2} = \sqrt{(132.28)^2 + (265.416)^2} = 296.55 \; m/s$$

$\therefore$ Diagram power or energy supplied $= m\left[\dfrac{V_1^2}{2} + \dfrac{V_{r2}^2 - V_{r1}^2}{2}\right]$

$$= \frac{m}{2}\left[2V_1^2 - V_{r1}^2\right] \qquad (\because V_{r2} = V_1)$$

$$= \frac{11.667}{2}\left[2 \times (776.026)^2 - (296.55)^2\right] = 6.513 \times 10^6 W$$

$$= \mathbf{6.513 \text{ MW}}$$ **Ans**

Problem 6.15 *In a certain stage of a reaction turbine, the steam leaves the fixed blade at a pressure of 2 bar 0.95 dry and a velocity of 160 m/s. The blades are 22 mm high and the discharge angle for both the blades is 20°. The ratio of the axial velocity of flow to the blade velocity is 0.75 at inlet and 0.8 at exit from moving blades. If the turbine uses 5 kg/s. of steam with 6% tip leakage, calculate the mean blade diameter and power developed in the ring.*

Solution : *Given data :*

$p = 2 \; bar, \; x = 0.95, \; V_1 = 160 \; m/s, \; h = 22 \, mm = 0.022 \, m, \; \alpha = \phi = 20°,$

$\left(V_{f1} / V_b\right) = 0.75, \; \left(V_{f2} / V_b\right) = 0.8 \; m = 5 \;\; kg/s$, *tip leakage* $= 6\%, \; D_m = ?, \; P = ?$

Actual mass flow rate, m_a = Mass flow rate – tip leakage

$$= 5(1 - 0.06) = 4.7 \; kg/s$$

Fig. 6.31 shows velocity diagram.

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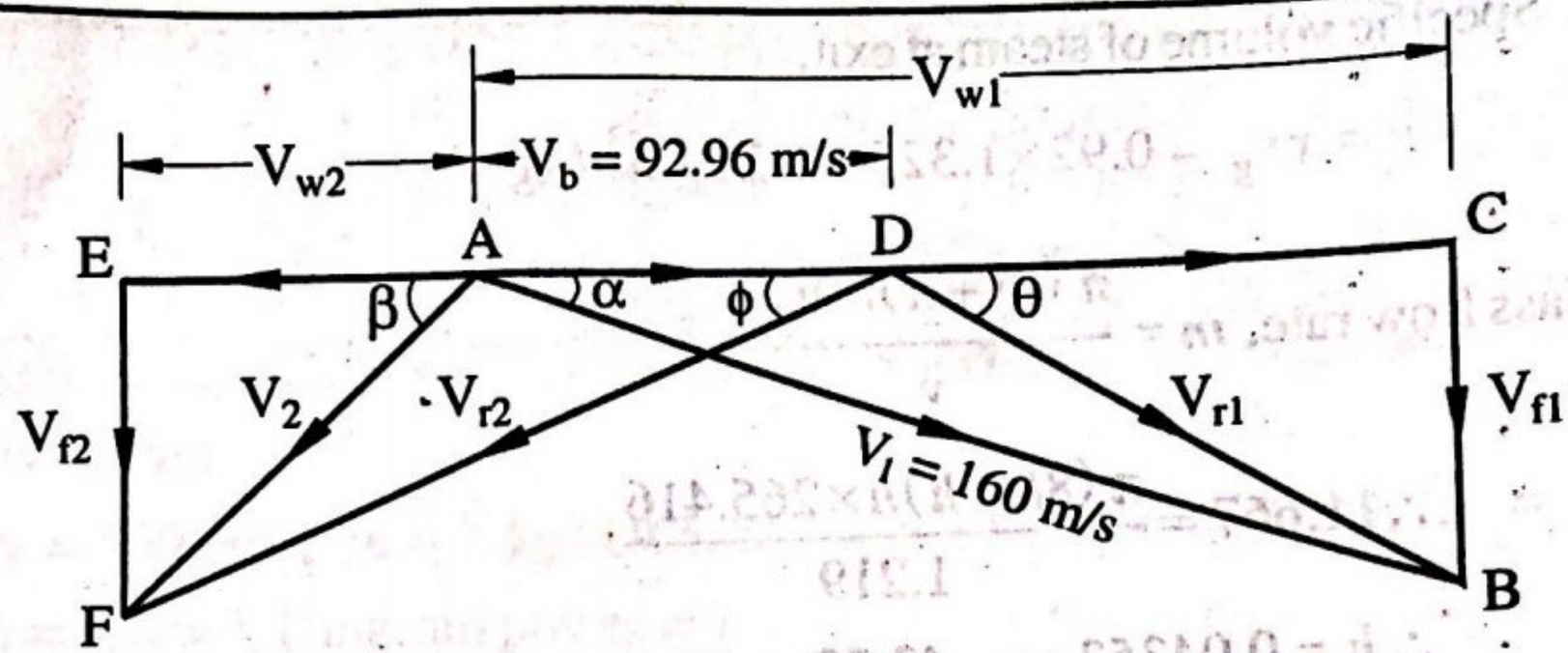


Fig. 6.31

$$V_{f1} = 0.75\ V_b = V_1 \sin\alpha$$

$$\therefore 0.75V_b = 160 \sin 20,\ \therefore\ V_b = 72.96\ m/s$$

$$\therefore V_{f1} = 0.75 \times 72.96 = 54.72\ m/s$$

$$V_{w1} = V_1 \cos\alpha = 160 \cos 20 = 150.35\ m/s$$

$$V_{f2} = 0.8\ V_b = 0.8 \times 72.96 = 58.368\ m/s$$

From triangle *DFE*,

$$ED = \frac{EF}{\tan\phi} = \frac{V_{f2}}{\tan\phi} = \frac{58.368}{\tan 20} = 160.365\ m/s$$

$$V_{w2} = EA = ED - AD = ED - V_b = 160.365 - 72.96 = 87.405\ m/s$$

$$\text{Average flow velocity, } V_f = \frac{V_{f1} + V_{f2}}{2} = \frac{54.72 + 58.368}{2} = 56.544\ m/s$$

From steam tables at $p = 2$ bar pressure, $v_g = 0.885\ m^3/kg$

$$\text{Specific volume of steam, } v = xv_g = 0.95 \times 0.885 = 0.84075\ m^3/kg$$

$$\text{Mass flow rate of steam, } m = \frac{\pi(D+h)h\,V_f}{v}$$

$$\therefore 5 = \frac{\pi(D + 0.022)0.022 \times 56.544}{0.84075}$$

$$\therefore D = 1.05367\ m$$

$$\therefore \text{ Mean drum diameter, } D_m = D + h = 1.05367 + 0.022$$

$$= \mathbf{1.0757\ m}$$

Ans

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Power developed, $P = \dfrac{m_a \times (V_{w1} + V_{w2}) V_b}{1000}$

$$= \frac{4.7 \times (150.35 + 87.405) \times 72.96}{1000}$$

$$= \mathbf{81.529\ kW}$$ **Ans**

Problem 6.16 *A Parson's turbine develops 1250 kW at 500 rpm and consumes 8.2 kg of steam per kWh, steam is supplied at 12 bar and 280°C and the isentropic efficiency of expansion is 86%. The blade angles are 37° and 20° at inlet and outlet respectively. Find the blade height and the drum diameter at a stage where the pressure is 1.5 bar. Drum diameter to blade height ratio of 12 is recommended. Determine also the power developed at that stage.*

Solution : Given data

$P = 1250\ kW,\ N = 500\ rpm,\ m = 8.2\ kg/kWh,\ p_1 = 12\ bar,\ T_1 = 280°C$

$\eta_{isen} = 86\%,\ \theta = \beta = 37°,\ \alpha = \phi = 20,\ p_2 = 1.5\ bar,\ D/h = 12,\ h = ?,\ D = ?,\ P' = ?$

Mass flow rate of steam = Steam consumption in kg/kWh × power in kW

$$= 8.2 \times 1250 = 10250\ kg/h$$

$$= \frac{10250}{3600} = 2.847\ kg/s$$

From Mollier's diagram for isentropic expansion $(1-2')$ as shown in Fig. 6.32, we get, $h_1 = 3030\ kJ/kg,\ h_2' = 2620\ kJ/kg$

Isentropic efficiency $\eta_{isen} = \dfrac{h_1 - h_2}{h_1 - h_2'} = 0.86 \equiv \dfrac{3030 - h_2}{3030 - 2620}$

$$\therefore h_2 = 2677.4\ kJ/kg$$

Fix state 2 on Mollier's diagram by drawing horizontal line at $h_2 \equiv 2677\ kJ/kg$ till it cuts the pressure line at $p_2 = 1.5\ bar$

On reading, $x_2 = 0.99$

From steam tables at $p_2 = 1.5\ bar$, $v_{g2} = 1.159\ m^3/kg$

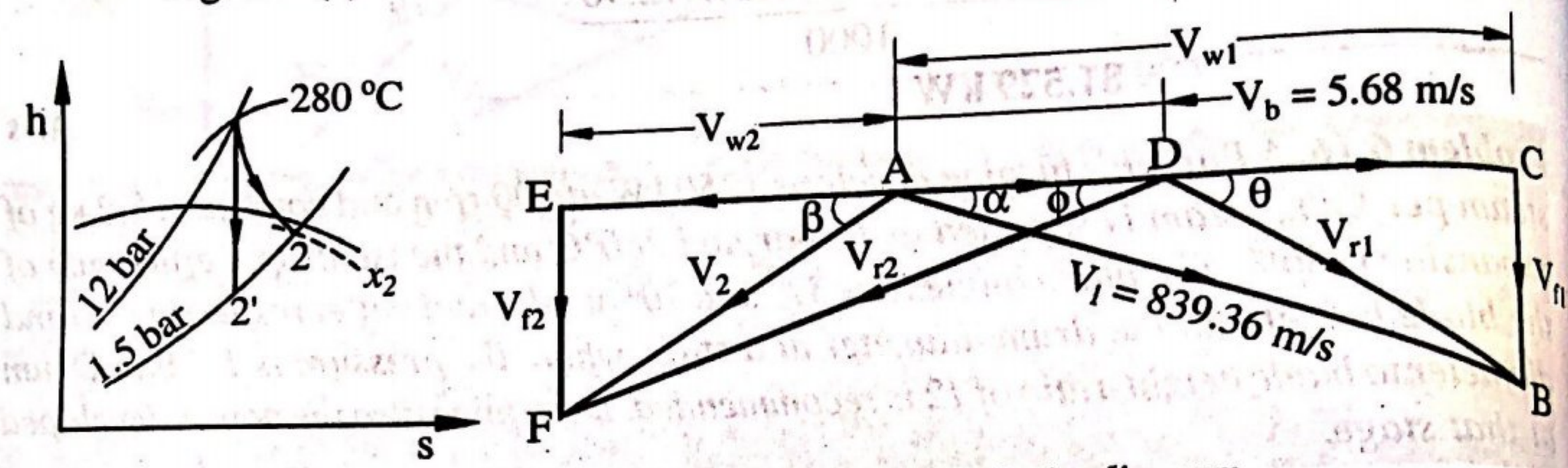
$\therefore$ Specific volume of steam, $v = x_2 v_{g2} = 0.99 \times 1.159 = 1.1474\ m^3/kg$

Inlet velocity of steam to turbine is given by

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$$V_1 = 44.7\sqrt{h_1 - h_2} = 44.7\sqrt{3030 - 2677.4} = 839.36 \text{ m/s}$$

Fig. 6.32(a) shows (h-s) diagram and Fig. 6.32(b) shows velocity diagram



(a) (h - s) diagram *(b) Velocity diagram*

Fig. 6.32

Velocity of flow, $V_{f1} = V_1 \sin\alpha = 839.36 \sin 20° = V_f = 287.078 \; m/s$

Mass flow rate, $m = \dfrac{\pi (D+h) h \, V_f}{v}$

$$2.847 = \frac{\pi (12h + h) h \times 287.078}{1.1474}$$

$$\therefore h = 0.01669 \; m = \mathbf{16.69 \text{ mm}}$$ *Ans*

Drum diameter, $D = 12h = 12 \times 16.69 = \mathbf{200.30 \text{ mm}}$ *Ans*

Blade velocity, $V_b = \dfrac{\pi (D+h) N}{60} = \dfrac{\pi (0.20028 + 0.01669) \times 500}{60}$

$$\therefore V_b = 5.68 \; m/s$$

$$V_{w1} = V_1 \cos\alpha = 839.36 \cos 20 = 788.74 \; m/s$$

$$V_{w2} = V_{w1} - V_b = 788.74 - 5.68 = 783.06 \; m/s$$

Power developed at the stage,

$$p' = \frac{m (V_{w1} + V_{w2}) V_b}{1000}$$

$$= \frac{2.847 (788.74 + 783.06) 5.68}{1000}$$

$$= \mathbf{25.4175 \text{ kW}}$$

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Problem 6.17 *The blade angles of a Parsons turbine at inlet and outlet are 47° and 20° respectively. The rotor speed is 1500 rpm, and the mean blade ring diameter is 1 m. The steam enters the stage at 9 bar dry and saturated. The leakage loss is estimated to be 8%. Calculate the blade height for 180 kW to be developed in the stage. Determine also the pressure drop in ring assuming a stage efficiency of 0.78.*

Solution : Given data :

$\theta = \beta = 47°$, $\alpha = \phi = 20°$, $N = 1500$ *rpm*, $D_m = 1$ *m*, $p_1 = 9$ *bar*, dry saturated, leakage of steam = 8%, $P = 180$ kW, $\eta_{stage} = 0.78$, h = ?, $p_1 - p_2 = ?$.

Actual mass flow rate of steam,

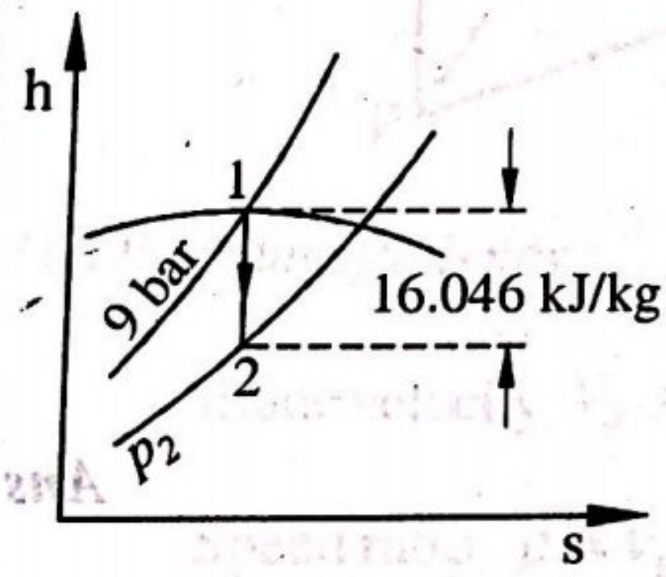
m = Theoretical mass flow rate x (1 – leakage factor)

$= m_t\,(1-0.08) = 0.92 m_t$

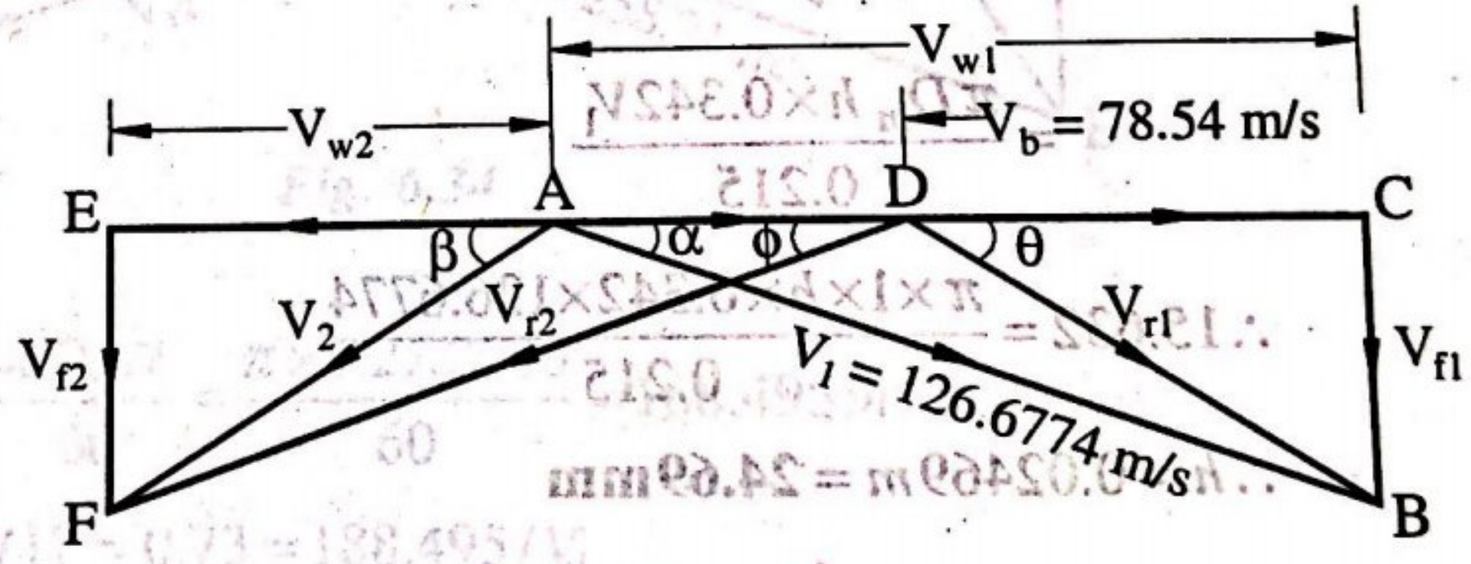
From steam tables at $p_1 = 9$ *bar*,

$v_g = v = 0.215\ m^3/kg$ ($\because$ steam is dry saturated)

Fig. 6.33(a) shows velocity diagram.



(a) (h - s) diagram



(b) Velocity diagram

Fig. 6.33

$$V_{w1} = V_1 \cos\alpha = V_1 \cos 20 = 0.939 V_1$$

$$V_{f1} = V_1 \sin\alpha = V_1 \sin 20 = 0.342 V_1$$

From triangle DBC, $DC = \dfrac{V_{f1}}{\tan\theta} = \dfrac{0.342 V_1}{\tan 47} = 0.319 V_1$

$$\therefore V_b = V_{w1} - DC = 0.939\, V_1 - 0.319\, V_1 = 0.62\, V_1$$

Also, $V_{w2} = AE = DC = 0.319 V_1$

$$\therefore V_{w1} + V_{w2} = 0.939 V_1 + 0.319 V_1 = 1.258 V_1$$

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Power developed, $P = \dfrac{m(V_{w1} + V_{w2})V_b}{1000}$

$$180 = \frac{0.92\, m_t \times 1.258 V_1 \times 0.62 V_1}{1000}$$

$$m_t\ V_1^2 = 250848.98$$

Blade velocity, $V_b = \dfrac{\pi D_m N}{60} = \dfrac{\pi \times 1 \times 1500}{60} = 78.54\ m/s$

Since, $V_b = 0.62 V_1$; $78.54 = 0.62 V_1$

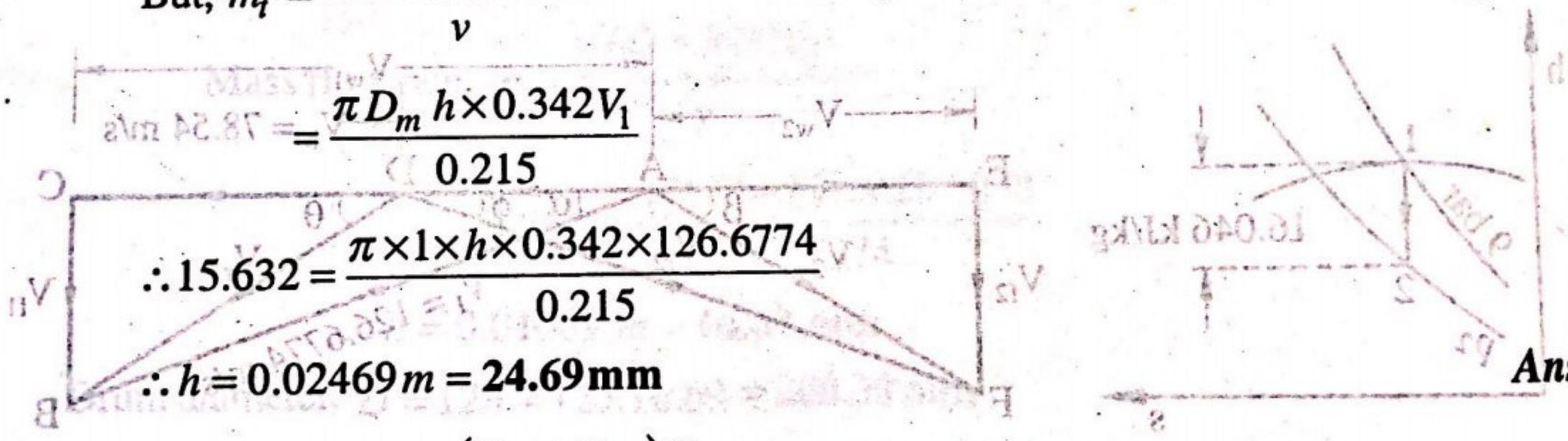
$$\therefore\ V_1 = 126.6774\, m/s$$

$$\therefore m_t\,(126.6774)^2 = 250848.98$$

$$\therefore m_t = 15.632\ kg/s$$

But, $m_t = \dfrac{\pi (D+h) h\, V_f}{v}$

$$= \frac{\pi D_m h \times 0.342 V_1}{0.215}$$



$$\therefore 15.632 = \frac{\pi \times 1 \times h \times 0.342 \times 126.6774}{0.215}$$

$$\therefore h = 0.02469\, m = \mathbf{24.69\,mm}$$ **Ans**

Stage efficiency, $\eta_{stage} = \dfrac{(V_{w1} + V_{w2})V_b}{\Delta h \times 1000}$

$$0.78 = \frac{1.258 V_1 \times 78.54}{\Delta h \times 1000} = \frac{1.258 \times 126.6774 \times 78.54}{\Delta h \times 1000}$$

$$\therefore \Delta h = 16.046\ kJ/kg$$

From Mollier's diagram as shown in Fig. 6.33(b)

$$p_2 \cong 8\, bar$$

$\therefore$ Pressure drop per stage, $p_1 - p_2 = 9 - 8 = \mathbf{1\,bar}$ **Ans**

Problem 6.18 *The following data relate to a stage of a Parson's reaction turbine. Mean rotor diameter = 1.2 m, speed ratio = 0.73, blade outlet angle = 20°, rotor speed = 3000 rpm.*

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Determine : (i) the diagram efficiency, (ii) the percentage increase in diagram efficiency and rotor speed if the rotor is designed to run at the best theoretical speed, the exit angle being 20°.

Solution : Given data

$D_m = 1.2\,m,\ \rho = \left(\frac{V_b}{V_1}\right) = 0.73,\ \alpha = \phi = 20°,\ N = 3000\ \ rpm,\ \eta_b = ?$ percentage increase in $\eta_b = ?, N = ?$

Fig. 6.34 shows velocity diagram

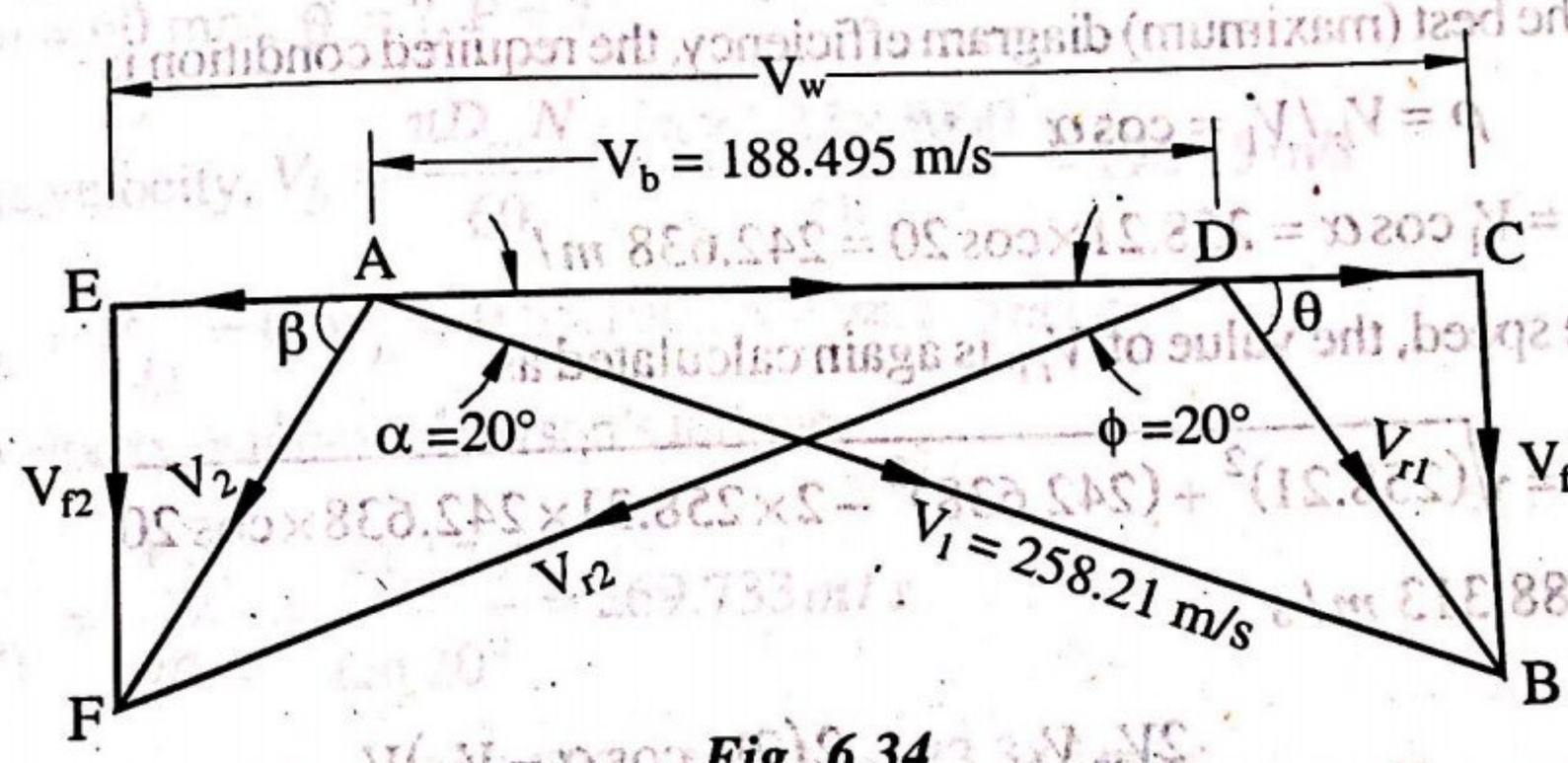


Fig. 6.34

(i) Diagram efficiency :

Blade velocity, $V_b = \frac{\pi D_m N}{60} = \frac{\pi \times 1.2 \times 3000}{60} = 188.495\,m/s$

Speed ratio, $\rho = V_b / V_1 = 0.73 = 188.495 / V_1$

$\therefore V_1 = 258.21\,m/s$

From the velocity triangle ADB, $V_{r1}^2 = V_1^2 + V_b^2 - 2V_1 V_b \cos\alpha$

$\therefore V_{r1} = \sqrt{(258.21)^2 + (188.495)^2 - 2(258.21)(188.495)\cos 20}$

$\therefore V_{r1} = 103.589\ m/s$

Work done per kg of steam $= V_w V_b = (2V_1 \cos\alpha - V_b) V_b$

$= (2 \times 258.21 \times \cos 20° - 188.495) \times 188.495 = 55941.75$ Nm

Energy supplied per kg of steam $= \frac{V_1^2}{2} + \frac{V_{r2}^2 - V_{r1}^2}{2}$

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$$= \frac{2V_1^2 - V_{r1}^2}{2} \quad (\because V_1 = V_{r2})$$

$$= \frac{2\times(258.21)^2 - (103.589)^2}{2} = 61307.064 Nm$$

$$\therefore \text{Diagram efficiency, } \eta_b = \frac{\text{Work done}}{\text{Energy supplied}} = \frac{55941.75}{61307.064}$$

$$= 0.91248 = \mathbf{91.248\%}$$ **Ans**

(ii) Percentage increase in diagram efficiency :

For the best (maximum) diagram efficiency, the required condition is

$$\rho = V_b / V_1 = \cos\alpha$$

$$\therefore V_b = V_1 \cos\alpha = 258.21\times\cos 20 = 242.638 \ m/s$$

For this blade speed, the value of V_{r1} is again calculated as

$$\therefore V_{r1} = \sqrt{(258.21)^2 + (242.638)^2 - 2\times 258.21\times 242.638\times\cos 20}$$

$$V_{r1} = 88.313 \ m/s$$

$$\text{Diagram efficiency, } \eta_b = \frac{2V_w V_b}{V_1^2 + V_{r2}^2 - V_{r1}^2} = \frac{2(2V_1\cos\alpha - V_b)V_b}{2V_1^2 - V_{r1}^2}$$

$$= \frac{2(2\times 258.21\cos 20 - 242.638)\times 242.638}{2\times(258.21)^2 - (88.313)^2}$$

$$= \frac{117746.43}{125545.622} = 0.93788 = 93.788\%$$

Percentage increase in diagram efficiency

$$= \frac{0.93788 - 0.91248}{0.91248}\times 100 = \mathbf{2.78\%}$$ **Ans**

The best theoretical speed of the rotor is given by

$$V_b = \frac{\pi D_m N}{60}, \quad \therefore N = \frac{60V_b}{\pi D_m} = \frac{60\times 242.638}{\pi\times 1.2}$$

$$\therefore N = \mathbf{3861.7 \ rpm}$$ **Ans**

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Problem 6.19 *The outlet angle of the blade of a Parson's turbine is 20° and the axial velocity of flow of steam is 0.5 times the mean blade velocity. If the mean diameter of the blade ring is 1.25 m and rotational speed is 3000 rpm, determine : (i) inlet angles of blades and (ii) power developed if dry-saturated steam at 5 bar passes through the blades. The blade height is 60 mm. Neglect the effect of blade thickness.*

[G.U., Dec. '06, 8 Marks]

Solution : Given data

$\alpha = \phi = 20^o$, $V_{f_1} = V_{f_2} = 0.5V_b$, $D_m = 1.25$ m, N = 3000 rpm,

$p = 5$ bar, $h = 60$ mm, $\theta = ?$, P = ?

$$\text{Mean blade velocity, } V_b = \frac{\pi D_m N}{60} = \frac{\pi \times 1.25 \times 3000}{60} = 196.35 \text{ m/s}$$

$$V_{f_1} = V_{f_2} = 0.5V_b = 0.5 \times 196.35 = 98.175 \, m/s$$

Referring velocity diagram of Parson's turbine,

$$V_{w_1} = \frac{V_{f_1}}{\tan \alpha} = \frac{98.175}{\tan 20^o} = 269.733 \, m/s$$

$$V_{w_2} = V_{w_1} - V_b = 269.733 - 196.35 = 73.383 \, m/s$$

$$\therefore V_w = V_{w_1} + V_{w_2} = 269.733 + 73.383 = 343.116 \text{ m/s}$$

$$\tan \theta = \frac{V_{f_1}}{V_{w_1} - V_b} = \frac{98.175}{73.383} = 1.3378$$

$\therefore \theta =$ **53.22°** ***Ans***

From steam tables at 5 bar pressure, specific volume of steam, $v = v_g = 0.375 \text{ m}^3/\text{kg}$

$$\text{Mass flow rate, } m = \frac{\pi D_m h V_f}{v} = \frac{\pi \times 1.25 \times 0.06 \times 98.175}{0.375} = 61.685 \text{ kg/s}$$

$$\text{Power developed, } P = \frac{m V_w V_b}{1000} = \frac{61.685 \times 343.116 \times 196.35}{1000} = \textbf{4155.77 kW} \quad \textit{Ans}$$

Problem 6.20 *The velocity of steam at inlet to a simple impulse turbine is 1000 m/s and nozzle angle is 20°. The mean blade speed is 400 m/s and blades are symmetrical. The mass flow rate is 0.75 kg/s. The friction effects on blades are negligible, estimate :*

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(i) blade angles, (ii) the tangential force on blades and (iii) axial thrust. **[G.U., Dec. 05, 8 Marks]**

Solution : Given data

$V_1 = 1000$ m/s, $\alpha = 20°$, $V_b = 400$ m/s, $\theta = \phi$, m = 0.75 kg/s, $\theta = ?$, $\phi = ?$, F = ?, $F_a = ?$

Refer velocity diagram for impulse turbine

(i) Blade angles :

$$V_{w_1} = V_1 \cos\alpha = 1000 \times \cos 20^\circ = 939.69\, m/s$$

$$V_{f_1} = V_1 \sin\alpha = 1000 \times \sin 20^\circ = 342.02\, m/s$$

$$DC = V_{w_1} - V_b = 939.69 - 400 = 539.60 \text{ m/s}$$

$$\therefore V_{r_1} = \sqrt{DC^2 + V_{f_1}^{\ 2}} = \sqrt{(539.69)^2 + (342.02)^2} = 638.94 \text{ m/s}$$

$$\tan\theta = \frac{V_{f_1}}{DC} = \frac{342.02}{539.69} = 0.6337$$

$\therefore$ Blade angles, $\theta = \phi = \mathbf{32.36^\circ}$ ***Ans***

(ii) The tangential force on blades :

The friction effects on blades are negligible,

$$\therefore V_{r_1} = V_{r_2} = 638.94 \text{ m/s}$$

$$V_{f_2} = V_{r_2} \sin\phi = 638.94 \sin 32.36^\circ = 341.984 \text{ m/s}$$

$$V_{w_2} = DE - AD = V_{r_2} \cos\phi - V_b = 638.94 \cos 32.36^\circ - 400 = 139.713 \text{ m/s}$$

Tangential force, $F = m(V_{w_1} + V_{w_2})$

$$= 0.75 \times (939.69 + 139.713) = \mathbf{809.55\ N}$$ ***Ans***

(iii) Axial thrust :

$$F_a = m(V_{f_1} - V_{f_2}) = 0.75(342.02 - 341.984) = \mathbf{0.027\ N}$$ ***Ans***

Problem 6.21 *In a simple impulse turbine, steam leaves the nozzle with a velocity of 900 m/s at an angle of 20° to the plane of rotation. The mean blade velocity is 60% of velocity for maximum efficiency. If the diagram efficiency is 70% and axial thrust is 40 N per kg of steam, calculate : (i) the blade angles, (ii) the heat lost in friction in kJ/kg, and (iii) power developed for a mass flow rate of 10 kg/s.* **[G.U., Dec. '05, 8 Marks]**

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Solution : Given data

$V_1 = 900$ m/s, $\alpha = 20^\circ$, $V_b = 0.6(V_b)_{opt}$, $\eta_b = 70\%$, $F_a = 40$ N,

$\theta = ?$, $\phi = ?$, Heat lost = ?, $P = ?$, $m = 10$ kg/s

Refer velocity diagram for impulse turbine

(i) The blade angles :

For maximum efficiency, $\rho = \left(\frac{V_b}{V_1}\right)_{opt} = \frac{\cos\alpha}{2} = \frac{\cos 20^\circ}{2}$

$\therefore \left(\frac{V_b}{V_1}\right)_{opt} = 0.4698$; $(V_b)_{opt} = 0.4698 \times V_1$

$\therefore (V_b)_{opt} = 0.4698 \times 900 = 422.82\, m/s$

$\therefore (V_b) = 0.6 \times 422.82 = 253.692\, m/s$

$V_{w_1} = V_1 \cos\alpha = 900 \times \cos 20^\circ = 845.723\, m/s$

$V_{f_1} = V_1 \sin\alpha = 900 \times \sin 20^\circ = 307.818\, m/s$

$DC = V_{w_1} - V_b = 845.723 - 253.692 = 592.031$ m/s

$V_{r_1} = \sqrt{DC^2 + V_{f_1}^2} = \sqrt{(592.031)^2 + (307.818)^2} = 667.272$ m/s

Diagram efficiency, $\eta_b = \frac{2(V_{w_1} + V_{w_2})V_b}{V_1^2}$

$\therefore 0.7 = \frac{2(845.723 + V_{w_2})253.692}{(900)^2}$

$\therefore V_{w_2} = 271.774$ m/s

Axial thrust per kg of steam, $F_a = V_{f_1} - V_{f_2}$

$\therefore 40 = 307.818 - V_{f_2}$ $\therefore V_{f_2} = 267.818$ m/s

$DE = V_{w_2} + V_b = 271.774 + 253.692 = 525.466\, m/s$

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$$\tan\theta = \frac{V_{f_1}}{DC} = \frac{307.818}{592.031} = 0.5199$$

$\therefore$ Inlet angle of moving blades, $\theta = \mathbf{27.47^o}$ ***Ans***

$$\tan\phi = \frac{V_{f_2}}{DE} = \frac{267.818}{525.466} = 0.5097$$

$\therefore$ Exit angle of moving blade, $\phi = \mathbf{27^o}$ ***Ans***

(ii) The heat lost in friction :

$$V_{r_2} = \sqrt{DE^2 + V_{f_2}^2} = \sqrt{(525.466)^2 + (267.818)^2} = 589.78\, m/s$$

$$\text{The heat lost in friction per kg of steam} = \frac{V_{r_1}^2 - V_{r_2}^2}{2\times 1000}$$

$$= \frac{(667.272)^2 - (589.78)^2}{2\times 1000} = \mathbf{48.705\ kJ/kg}$$ ***Ans***

(iii) Power developed :

$$P = \frac{m(V_{w_1} + V_{w_2})V_b}{1000} = \frac{10(845.723 + 271.774)\times 253.692}{1000}$$

$$= \mathbf{2835\ kW}$$ ***Ans***

6.12 Reheat factor

Fig 6.35 shows the expansion of steam through a multi-stage turbine (say three-stage turbine)

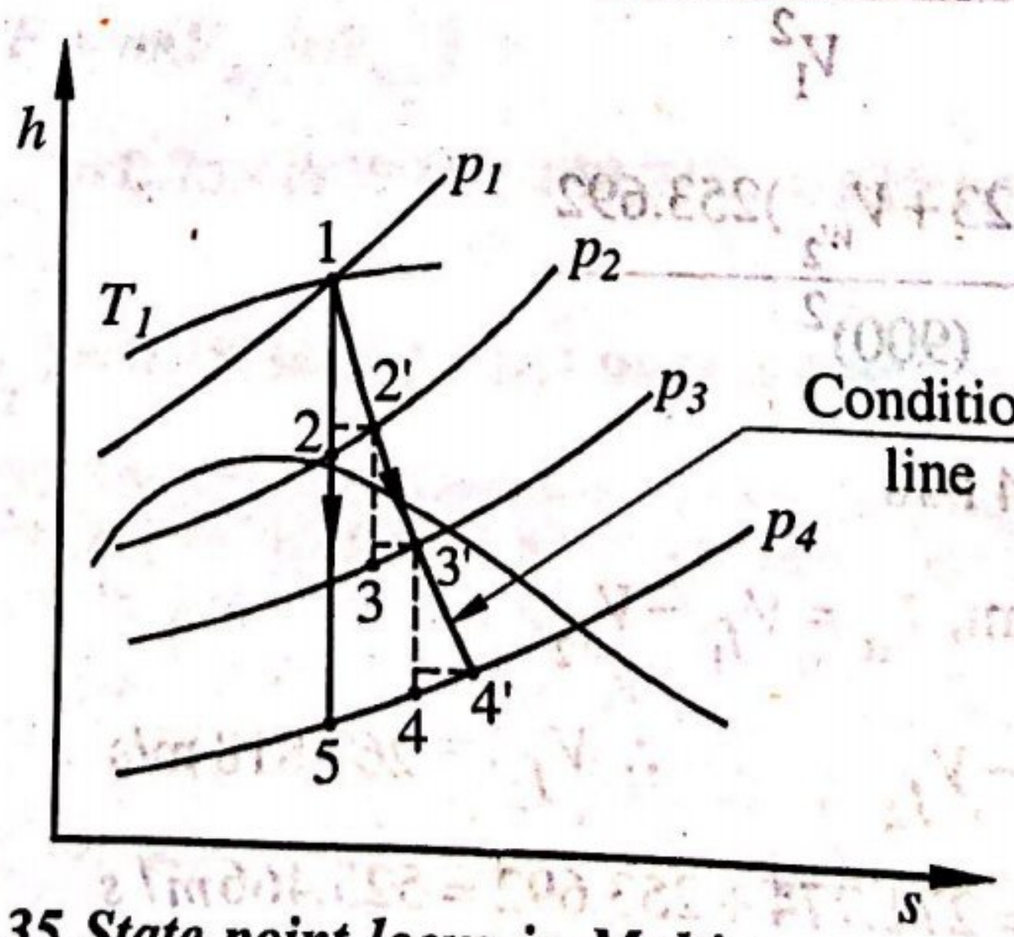


Fig. 6.35 State point locus in Multistage steam turbine

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Let p_1 = initial pressure of the steam,
p_2 = pressure of steam leaving the first stage,
p_3 = pressure of steam leaving the second stage,
p_4 = final pressure of the steam,
T_1 = initial temperature of the steam,
η_s = stage efficiency for each stage of the turbine.

Now let us draw the expansion of steam in three stages on a Mollier chart as shown in Fig. 6.35 as discussed below.

First of all, locate the point 1 with the help of initial pressure p_1 and temperature T_1 of the steam. 1-2 shows the isentropic expansion of steam in the first stage. But due to losses the actual state point of the steam after expansion in the first stage is $2'$ and not 2. Thus the actual heat drop is 1-$2'$ and not 1-2 so, we have

$$h_1 - h_2' = \eta_s (h_1 - h_2)$$

and point $2'$ is thus set down with the h_2' value on p_2 line.

When point $2'$ is known, isentropic expansion is drawn giving $(h_2' - h_3)$ and h_3' will be calculated from the relation

$$(h_2' - h_3') = \eta_s (h_2' - h_3)$$

and the point $3'$ is set down. Similarly, other points in the succeeding stages are set down.

The points 1, $2'$, $3'$ and $4'$ represent the condition of steam at the blade outlets of the three stages. *1-5 is the Rankine heat drop or isentropic heat drop for the initial and final pressure*. The curve joining the points 1, $2'$, $3'$, $4'$ is called the condition line of the expansion of steam because these points represent the final condition of steam at the end of each stage. *The sum of the isentropic heat drops 1-2, $2'$-3, $3'$-4, called cumulative heat drop is represented by* Δh_c. The cumulative heat drop is always greater than direct isentropic or Rankine heat drop 1-5 because the constant pressure line diverge as we move from left to right on the *h-s* diagram.

As discussed above, there are various losses like fluid and blade friction, leakages, shock, etc. in the stage and the portion of the available energy not converted to work and remaining in the fluid is termed as "*reheat*".

Reheat factor (RF) *is defined as the ratio of cumulative heat drop,* Δh_c *to the direct isentropic or Rankine heat drop,* Δh_{isen}. Mathematically

$$\text{RF} = \frac{\text{Cumulative heat drop } (\Delta h_c)}{\text{Rankine heat drop } (\Delta h_{isen})}$$

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Referring to Fig. 6.35

$$RF = \frac{\Delta h_{1-2} + \Delta h_{2'-3} + \Delta h_{3'-4}}{\Delta h_{1-5}} \quad ...(6.23)$$

The reheat factor depends on the turbine stage efficiency, the condition of steam at entry and the exit pressure. *The value of RF is always greater than unity because cumulative heat drop is always greater than Rankine heat drop*. The reheat factor is greater if the number of stages are large for a given pressure range and lower the stage efficiency. The value of RF usually varies between 1.02 to 1.06. Note that lower value of reheat factor is desirable.

6.13 Internal and other efficiencies

(1) Internal efficiency :

The internal efficiency of the turbine is defined as the ratio of total useful heat drop or internal work done in the turbine to the Rankine heat drop.

Mathematically

$$\eta_i = \frac{\text{Total useful heat drop}\,(\Delta h_i)}{\text{Rankine heat drop}\,(\Delta h_{isen})} \quad ...(6.24)$$

$$= \frac{\Delta h_{1-2'} + \Delta h_{2'-3'} + \Delta h_{3'-4'}}{\Delta h_{1-5}}$$

If the stage efficiency η_s is the same for all stages or if η_s be the mean stage efficiency, then we have

$$\eta_s = \frac{\Delta h_{1-2'}}{\Delta h_{1-2}} = \frac{\Delta h_{2'-3'}}{\Delta h_{2'-3}} = \frac{\Delta h_{3'-4'}}{\Delta h_{3'-4}} = \frac{\Delta h_{1-2'} + \Delta h_{2'-3'} + \Delta h_{3'-4'}}{\Delta h_{1-2} + \Delta h_{2'-3} + \Delta h_{3'-4}}$$

$$= \frac{\Delta h_i}{\Delta h_c}$$

$$\therefore \Delta h_i = \eta_s \Delta h_c$$

But $RF = \dfrac{\Delta h_c}{\Delta h_{isen}}$

$$\therefore \Delta h_i = (RF)\eta_s \Delta h_{isen}$$

Substituting this value of Δh_i in equation (6.24)

$$\eta_i = (RF) \times \eta_s \quad ...(6.25)$$

The internal power of a turbine is

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$$IP = \dot{m}_i \Delta h_i \text{ kW}$$

(2) Isentropic efficiency :

It is defined as the ratio of actual heat drop to isentropic heat drop in whole turbine

$$\therefore \eta_{isen} = \frac{\Delta h_{act}}{\Delta h_{isen}}$$

If RF = 1, $\eta_{isen} = \eta_i = \eta_s$

(3) Overall thermal efficiency of turbo-generator :

It is the ratio of electrical generator output to actual heat supplied. Mathematically

$$\therefore \eta_{oth} = \frac{\text{Generator output}}{\text{Actual heat supplied}}$$

Neglecting pump work,

$$\eta_{oth} = (\eta_{th})_{isen} \times \eta_{boiler} \times \eta_i \times \eta_m \times \eta_g$$

(4) Efficiency ratio :

It is defined as the ratio of the total work produced at the driving end of the shaft per kg of steam to the total isentropic heat drop across the turbine. Mathematically, efficiency ratio,

$$\text{E.R.} = \frac{\text{Total work produced at the driving end of shaft}}{\text{Total isentropic heat drop}}$$

$$= \frac{\text{Total useful heat drop}}{\text{Total isentropic heat drop}}$$

Problem 6.22 *Steam at 25 bar 450°C expands in a steam turbine to 0.1 bar. There are four stages in the turbine and the total enthalpy drop is divided equally among the stages. The stage efficiency is 76% and it is the same in all the stages. Determine the interstage pressures, the reheat factor and the turbine internal efficiency.*

Solution : Given data (Refer Fig. 6.36)

$p_1 = 25$ bar, $T_1 = 450°C$, $p_5 = 0.1$ bar, $\eta_s = 76\%$

From Mollier chart,

$$h_1 = 3350 \text{ kJ/kg}, \quad h_6 = 2280 \text{ kJ/kg}$$

$$\therefore h_1 - h_6 = 3350 - 2280 = 1070 \text{ kJ/kg}$$

Since total enthalpy drop is divided equally among the four stages,

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$$\therefore h_1 - h_2 = \frac{h_1 - h_6}{4} = \frac{1070}{4} = 267.5 \text{kJ/kg}$$

$$\therefore 3350 - h_2 = 267.5, \ \therefore h_2 = 3082.5 \ kJ/kg$$

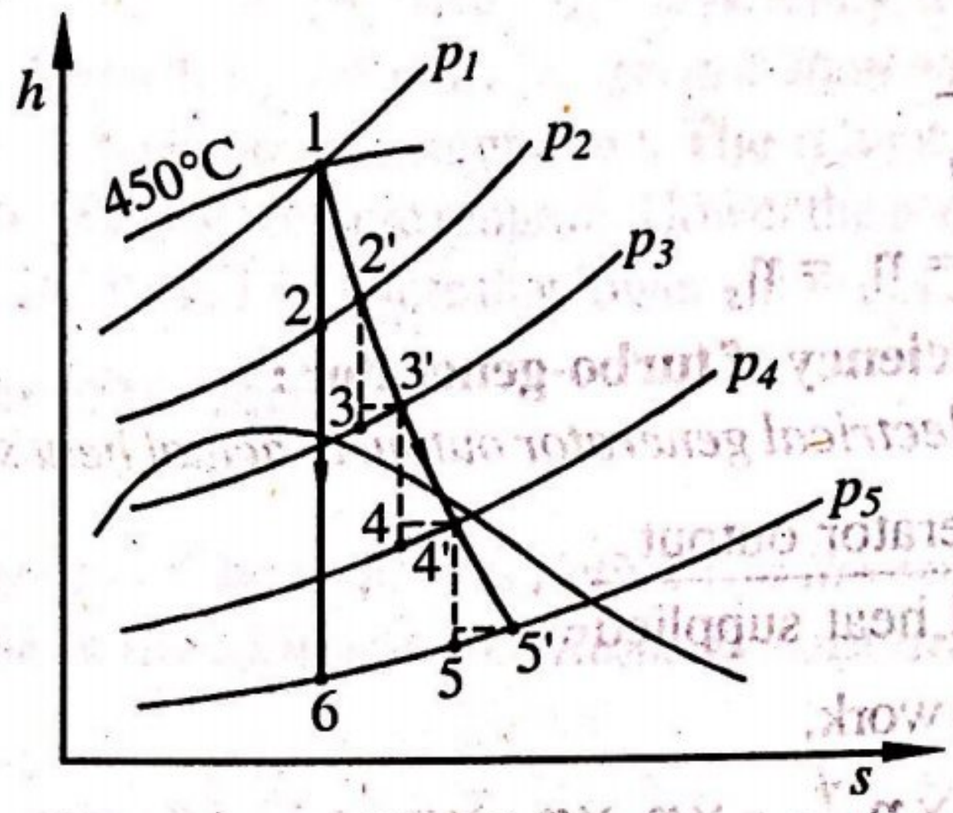


Fig. 6.36

The interstage pressures p_2, p_3 and p_4 as read from the Mollier chart are **10 bar, 3 bar and 0.75 bar** respectively. *Ans*

$$h_1 - h_2' = \eta_s (h_1 - h_2) = 0.76 \times 267.5 = 203.3 \text{ kJ/kg}$$

$$\therefore h_2' = 3146.7 \text{ kJ/kg}$$

$$h_3 = 2850 \text{ kJ/kg}$$

$$\therefore h_2' - h_3 = 3146.7 - 2850 = 296.7 \ kJ/kg$$

$$\therefore h_2' - h_3' = 0.76 \times 296.7 = 225.492 \text{ kJ/kg}$$

$$\therefore h_3' = 2921.208 \text{ kJ/kg}$$

$$h_4 = 2665 \text{ kJ/kg}$$

$$\therefore h_3' - h_4 = 2921.208 - 2665 = 256.208 \text{ kJ/kg}$$

$$\therefore h_3' - h_4' = 0.76 \times 256.208 = 194.72 \text{ kJ/kg}$$

$$\therefore h_4' = 2726.488 \text{ kJ/kg}$$

$$h_5 = 2435 \text{ kJ/kg}$$

$$\therefore h_4' - h_5 = 2726.488 - 2435 = 291.488 \text{ kJ/kg}$$

$$\therefore h_4' - h_5' = 0.76 \times 291.488 = 221.53 \text{ kJ/kg}$$

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$$\therefore h_5' = 2504.957 \text{ kJ/kg}$$

$$x_5' = 0.965$$

$$\text{Reheat factor, RF} = \frac{(h_1 - h_2) + (h_2' - h_3) + (h_3' - h_4) + (h_4' - h_5)}{h_1 - h_6}$$

$$= \frac{267.5 + 296.7 + 256.208 + 291.488}{1070}$$

$$\text{RF} = \mathbf{1.039}$$ **Ans**

$$\text{Internal efficiency, } \eta_i = \frac{h_1 - h_5'}{h_1 - h_6} = \frac{3350 - 2504.957}{1070} = 0.7897 = \mathbf{78.97\%}$$ **Ans**

$$\text{Alternatively, } \eta_i = \eta_s \times RF = 0.76 \times 1.039 = 0.7896 = \mathbf{78.96\ \%}$$ **Ans**

Problem 6.23 *A steam is supplied to three stage turbine at 30 bar and 400°C. Steam is exhausted at 0.05 bar. The interstage pressures are 6 bar and 1 bar. Stage efficiency for all stages is 76%. Assuming condition line to be straight, determine : (i) Rankine efficiency, (ii) quality of steam leaving each stage, (iii) reheat factor, (iv) work done per kg of steam in each case, and (v) overall efficiency.* ***[May '15, same as Sep. '13]***

Solution : Given data (Refer Fig. 6.37)

$p_1 = 30$ bar, $T_1 = 400°C$, $p_4 = 0.05$ bar, $p_2 = 6$ bar, $p_3 = 1$ bar, $\eta_s = 76\%$

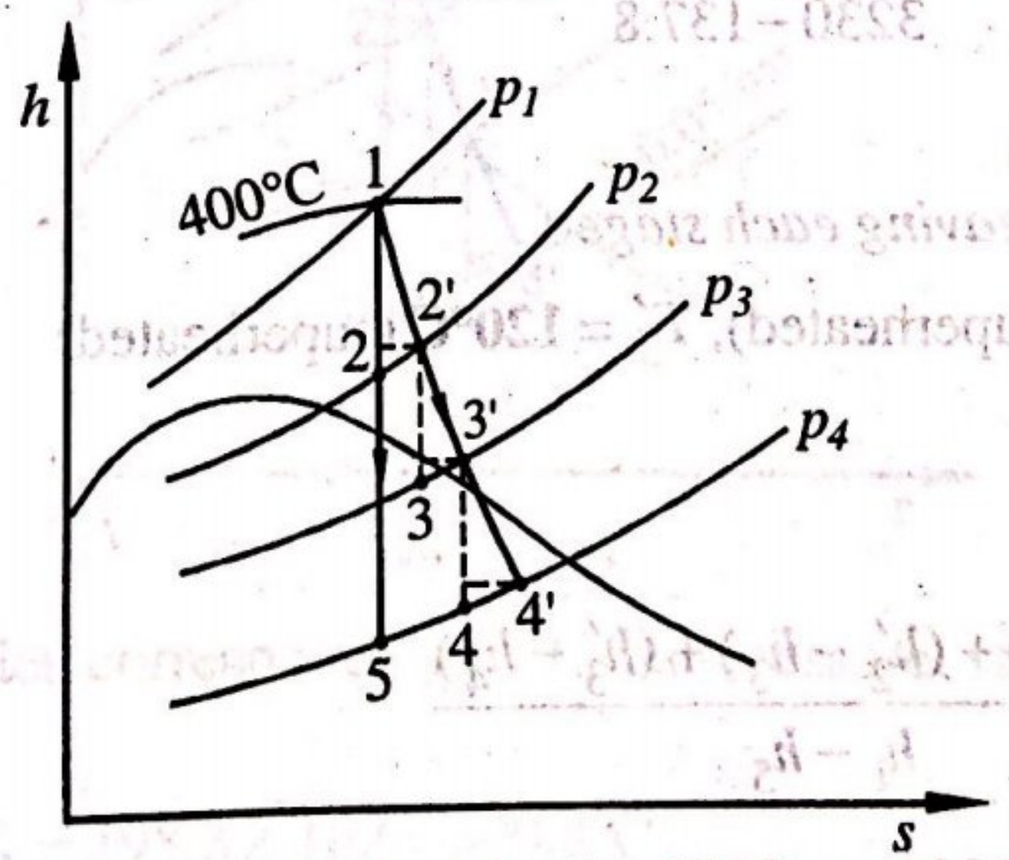


Fig. 6.37

(i) Rankine efficiency, η_R :

From Mollier's diagram

$h_1 = 3230$ kJ/kg, $h_5 = 2110$ kJ/kg

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Stage efficiency, $\eta_s = \dfrac{h_1 - h_4'}{h_1 - h_5}$

$$\therefore 0.76 = \frac{3230 - h_4'}{3230 - 2110}$$

$$\therefore h_4' = 2378.8 \text{ kJ/kg}$$

Fix point $4'$ on Mollier's diagram by drawing horizontal line at $h_4' = 2378.8$ kJ/kg which cuts $p_4 = 0.05$ bar line at $4'$.

Joint 1-$4'$ by a straight line since the condition line is a straight line. It cuts p_2 and p_3 lines at $2'$ and $3'$.

Draw isentropic lines 1-2, $2'$-3 and $3'$-4 and read :

$h_2' = 2960$ kJ/kg, $h_2 = 2830$ kJ/kg, $h_3' = 2710$ kJ/kg

$h_3 = 2510$ kJ/kg, $h_4 = 2270$ kJ/kg

From steam tables at 0.05 bar, $h_{f_5} = 137.8$ kJ/kg

Neglecting pump work, Rankine efficiency

$$\eta_R = \frac{h_1 - h_5}{h_1 - h_{f_5}} = \frac{3230 - 2110}{3230 - 137.8} = 0.3622$$

$$= \mathbf{36.22\ \%} \quad \textit{Ans}$$

(ii) Quality of steam leaving each stage :

$T_2' = \mathbf{255^oC}$ (superheated), $T_3' = \mathbf{120^oC}$ (superheated)

$x_4' = \mathbf{0.92}$ **dry** *Ans*

(iii) Reheat factor :

$$\text{RF} = \frac{(h_1 - h_2) + (h_2' - h_3) + (h_3' - h_4)}{h_1 - h_5}$$

$$= \frac{(3230 - 2830) + (2960 - 2510) + (2710 - 2270)}{3230 - 2110}$$

$$\text{RF} = \mathbf{1.152}$$

(iv) Work done in each stage : *Ans*

First stage : $w_1 = h_1 - h_2' = 3230 - 2960 = \mathbf{270\ kJ/kg}$ *Ans*

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Second stage : $w_2 = h_2' - h_3' = 2960 - 2710 = \mathbf{250\ kJ/kg}$ **Ans**

Third stage : $w_3 = h_3' - h_4' = 2710 - 2378.8 = \mathbf{331.2\ kJ/kg}$ **Ans**

(v) Overall efficiency :

$$\eta_o = \frac{h_1 - h_4'}{h_1 - h_{f_5}} = \frac{3230 - 2378.8}{3230 - 137.8} = 0.2753 = \mathbf{27.53\%}$$ **Ans**

Problem 6.24 *An impulse turbine installation, consisting of H.P., I.P., and L.P. turbines, is required to work with initial steam conditions of 15 bar with 102°C of superheat, and a condenser pressure of 0.05 bar. Allowing a reheat factor of 1.06 and a loss of available heat of 5 kJ/kg of steam, determine the heat units to be allocated to each turbine in order that the H.P. and I.P. may each developed 1/4 of the total power. Assume stage efficiencies of 0.76, 0.77, and 0.73 in the H.P., I.P. and L.P. respectively.*

Solution : Given data (Refer Fig. 6.38)

$p_1 = 15$ bar, Degree of superheat = 102°C, $p_4 = 0.05$ bar, RF = 1.06,

loss of heat = 5 kJ/kg

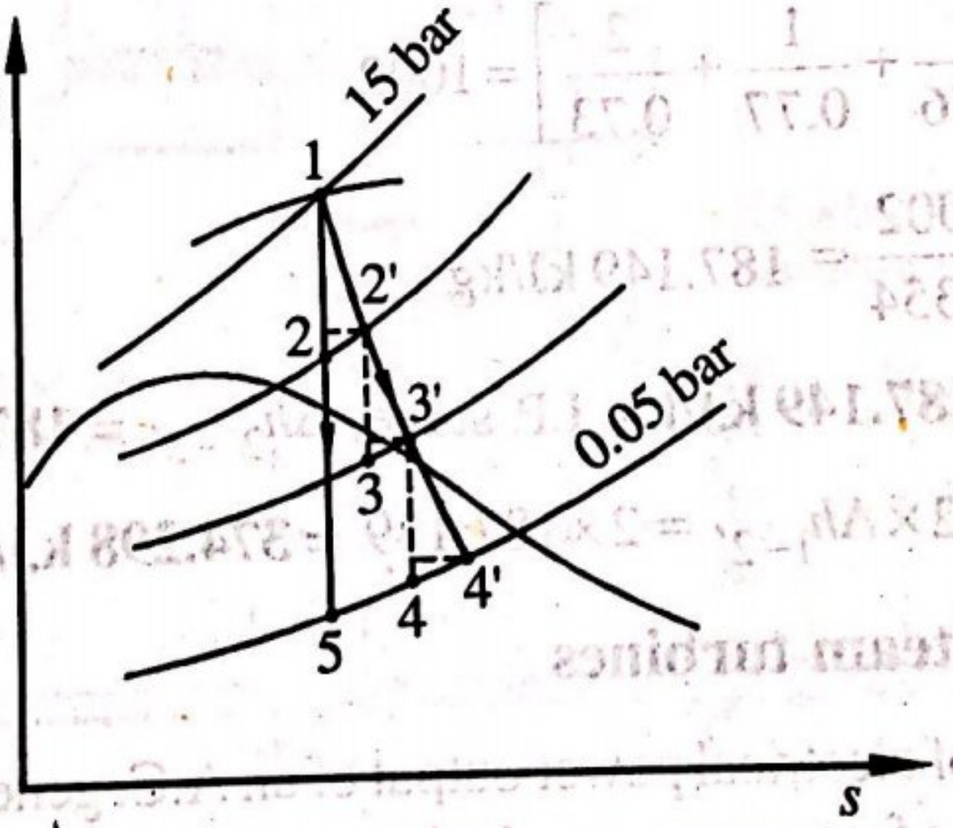


Fig. 6.38

From steam tables corresponding to 15 bar pressure, saturation temperature = 198.3°C

$\therefore$ Initial temperature, $T_1 = 198.3 + 102 = 300.3$°C

From Mollier chart : $h_1 = 3040$ kJ/kg, $h_5 = 2090$ kJ/kg

Total isentropic heat drop, $(h_1 - h_5) = 3040 - 2090 = 950$ kJ/kg

$$\text{RF} = \frac{(h_1 - h_2) + (h_2' - h_3) + (h_3' - h_4) + \text{loss of heat}}{h_1 - h_5}$$

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$$1.06 = \frac{(h_1 - h_2) + (h_2' - h_3) + (h_3' - h_4) + 5}{950}$$

$$\therefore (h_1 - h_2) + (h_2' - h_3) + (h_3' - h_4) = \Delta h_c = 950 \times 1.06 - 5 = 1002 \text{ kJ/kg}$$

Workdone in each turbine is $\Delta h_{1-2'}$, $\Delta h_{2'-3'}$ and $\Delta h_{3'-4'}$ and according to problem

$$\Delta h_{1-2'} = \Delta h_{2'-3'} = \frac{\Delta h_{1-4'}}{4} = \frac{\Delta h_{3'-4'}}{2}$$

Also, we have

$$\Delta h_{1-2} = \frac{\Delta h_{1-2'}}{\eta_s}$$

$$\therefore \frac{\Delta h_{1-2'}}{0.76} + \frac{\Delta h_{2'-3'}}{0.77} + \frac{\Delta h_{3'-4'}}{0.73} = 1002$$

$$\therefore \frac{\Delta h_{1-2'}}{0.76} + \frac{\Delta h_{1-2'}}{0.77} + \frac{2 \times \Delta h_{1-2'}}{0.73} = 1002$$

$$\therefore \Delta h_{1-2'} \left[\frac{1}{0.76} + \frac{1}{0.77} + \frac{2}{0.73}\right] = 1002$$

$$\therefore \Delta h_{1-2'} = \frac{1002}{5.354} = 187.149 \text{ kJ/kg}$$

H.P. stage, $\Delta h_{1-2'} = \mathbf{187.149\ kJ/kg}$, I.P. stage, $\Delta h_{2'-3'} = \mathbf{187.149\ kJ/kg}$

L.P. stage, $\Delta h_{3'-4'} = 2 \times \Delta h_{1-2'} = 2 \times 187.149 = \mathbf{374.298\ kJ/kg}$ ***Ans***

6.14 Governing of steam turbines [May '14, May '12]

The frequency of electrical power output of an A.C. generator is directly affected by the speed of turbine. The frequency must be kept within certain limits for the satisfactory operation of electrical appliances. This is done with the help of a governor which maintains the speed of the turbine almost constant irrespective of variations in the load on turbine. Hence, the function of the governor is to maintain the shaft speed constant as the load varies.

The principal methods of steam turbine governing are as follows :

(1) Throttle governing
(2) Nozzle governing
(3) By-pass governing
(4) Combined throttle and nozzle governing.
(5) Combined throttle and by-pass governing.

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6.14.1 Throttle governing

The object of throttle governing is to throttle the steam to a suitable pressure and allow required quantity of steam to flow through the turbine according to load for maintaining the constant speed of the turbine.

The arrangement of throttle governing is shown in Fig. 6.39. To start the turbine for full load running stop valve is opened. The operation of double beat throttle valve is carried out by an oil servomotor which is controlled by a centrifugal governor.

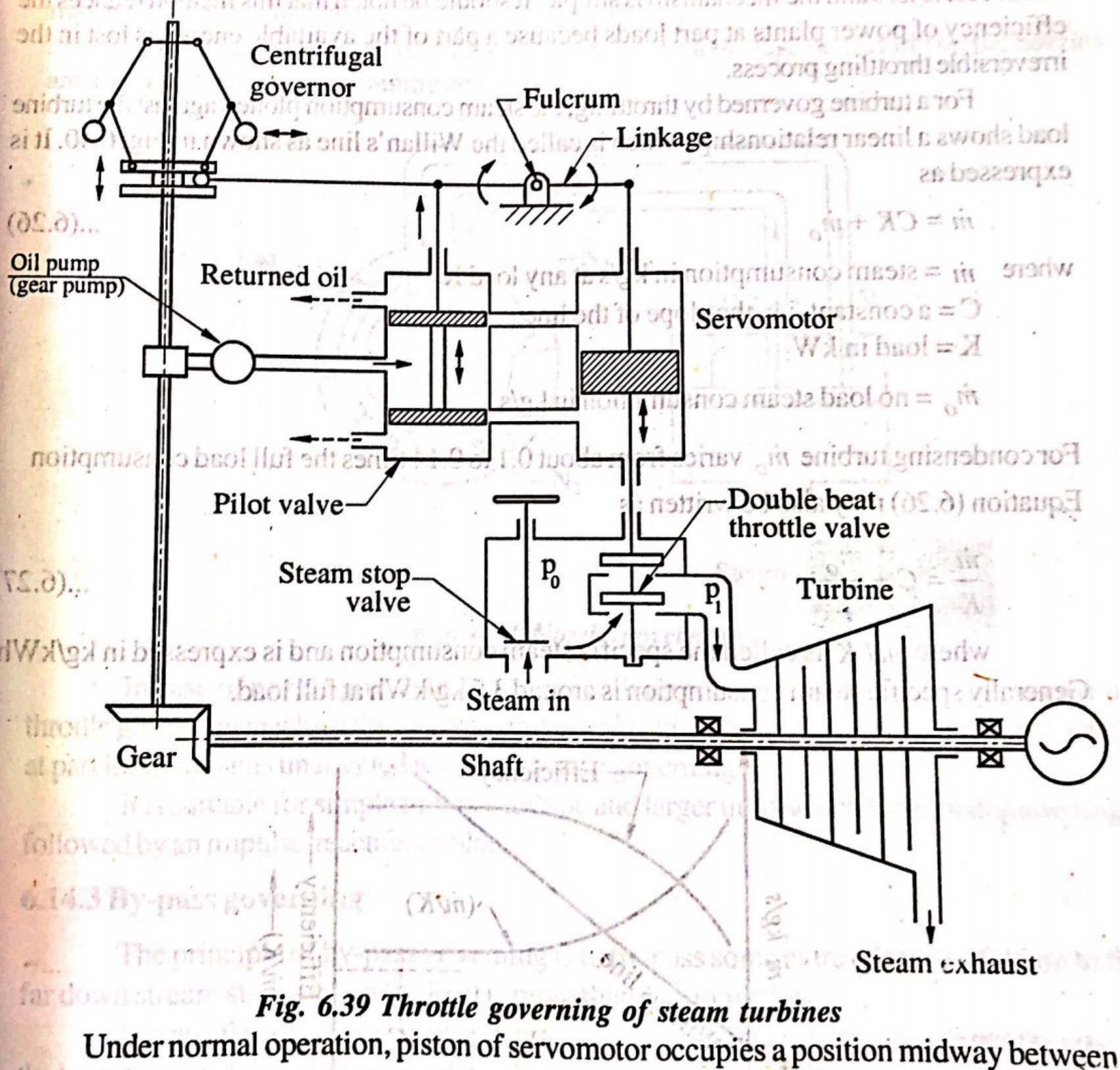


Fig. 6.39 Throttle governing of steam turbines

Under normal operation, piston of servomotor occupies a position midway between its travel in which both the inlet and exit ports of the pilot valve connecting it to the servomotor are closed. Any displacement of sleeve of centrifugal governor due to change in electrical load causes a displacement of the pilot valve. According to the displacement, the oil under pressure from the oil pump enters either upper or lower half of the chamber of servomotor. If oil enters

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the upper half portion of servomotor, the throttle valve starts closing reducing the quantity of steam flowing through the turbine thus reducing the power output of turbine till the speed is maintained fairly near nominal value. At the same time oil from bottom half of servomotor starts flowing out through the pilot valve port to drain. If oil under pressure enters the lower half chamber of servomotor, an exactly opposite process is affected opening the throttle valve till the speed is maintained.

Throttle governing is the most widely used particularly in small turbines, because its initial cost is less and the mechanism is simple. It should be noted that this method reduces the efficiency of power plants at part loads because a part of the available energy is lost in the irreversible throttling process.

For a turbine governed by throttling, the steam consumption plotted against the turbine load shows a linear relationship, which is called the Willan's line as shown in Fig. 6.40. It is expressed as

$$\dot{m} = CK + \dot{m}_o \qquad ...(6.26)$$

where $\dot{m}$ = steam consumption in kg/s at any load K
C = a constant, i.e. the slope of the line
K = load in kW
$\dot{m}_o$ = no load steam consumption in kg/s

For condensing turbine $\dot{m}_o$ varies from about 0.1 to 0.14 times the full load consumption

Equation (6.26) may also be written as

$$\frac{\dot{m}}{K} = C + \frac{\dot{m}_o}{K} \qquad ...(6.27)$$

where $\dot{m}/K$ is called the specific steam consumption and is expressed in kg/kWh. Generally specific steam consumption is around 3.5 kg/kWh at full load.

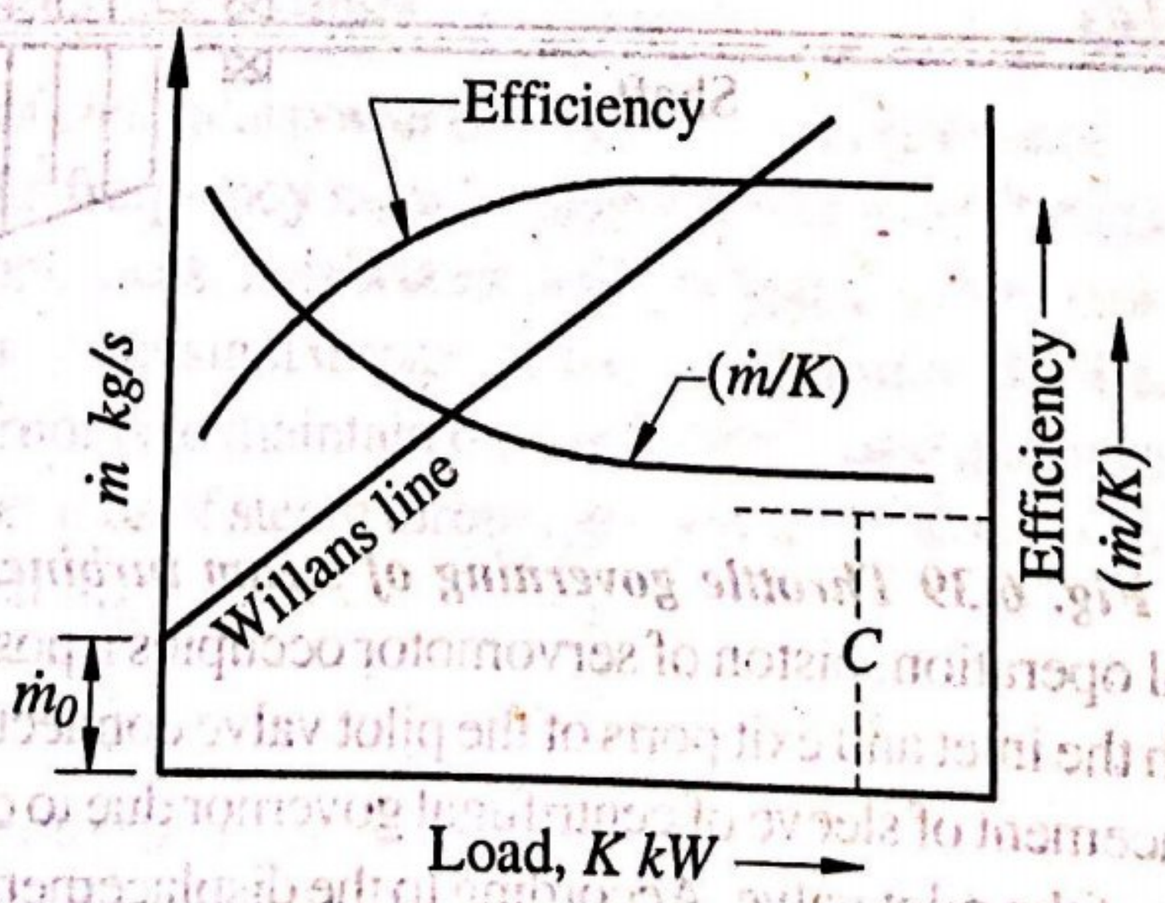


Fig. 6.40 Throttle governing characteristics

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6.14.2 Nozzle governing

[Nov. '14, May '12]

In this method, the mass flow rate of steam supplied to the turbine is only controlled. It is more efficient than the throttle governing. In this method, the nozzles of the turbines are grouped in two, three or more groups and each group of nozzles is fed with steam controlled by valves. The arrangement of nozzle control governing is shown in Fig. 6.41.

Under full load conditions all the valves remain fully open. When the load on the turbine becomes more or less than the design value, the supply of steam to a group of nozzles may be varied accordingly so as to restore the original speed.

The nozzle control is necessarily restricted to the first stage of the turbine, the nozzles areas in the other stages remaining constant.

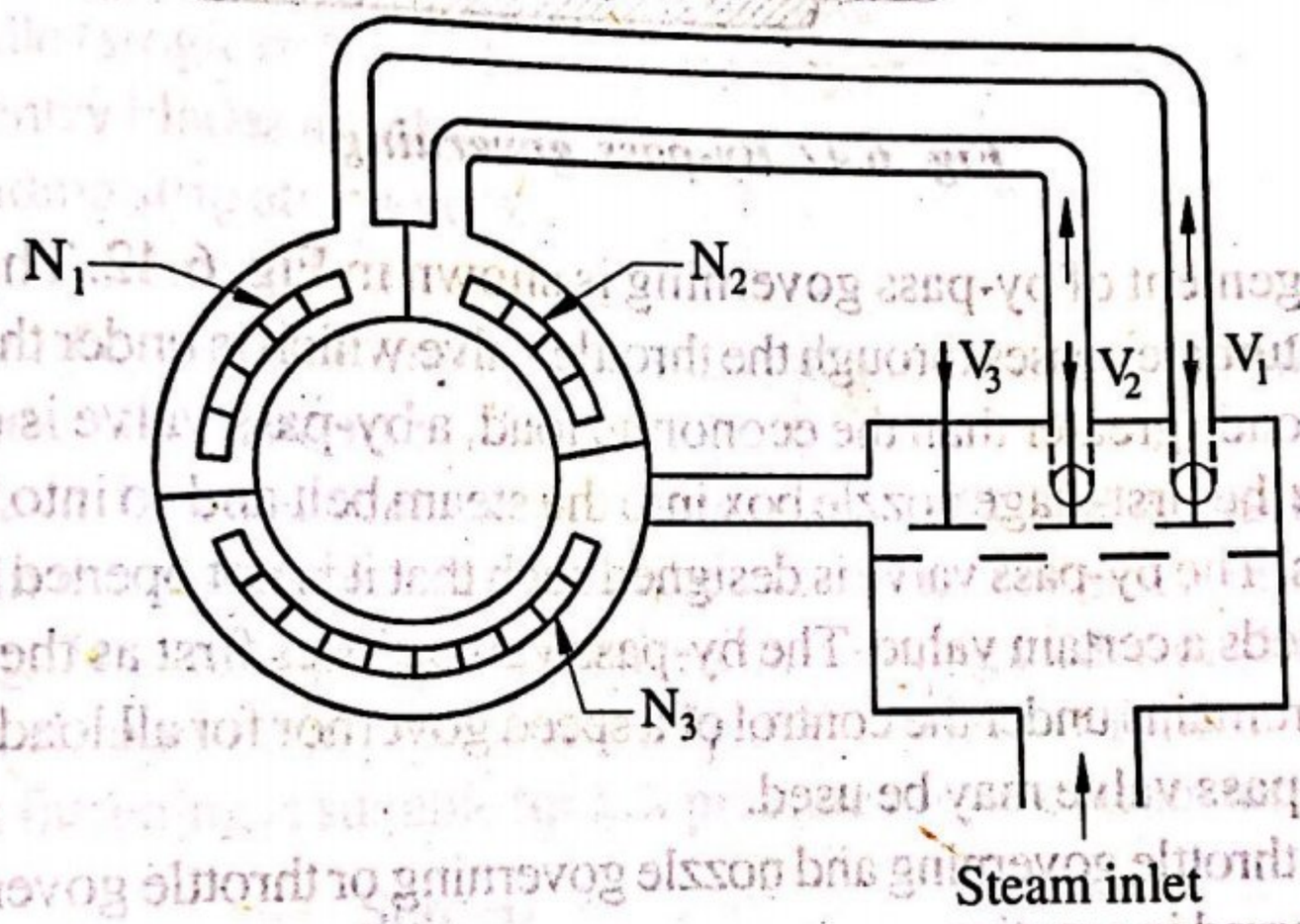


Fig. 6.41 Nozzle governing

In case of nozzle governing, the total enthalpy drop is available unlike the case of throttle governing in which the enthalpy drop is reduced at part loads. Due to this the efficiency at part loads remains unaffected in case of nozzle governing.

It is suitable for simple impulse turbine and larger units which have an impulse stage followed by an impulse-reaction turbine.

6.14.3 By-pass governing

The principle of by-pass governing is to by-pass some extra quantity of steam to the far down stream stages when the load is more than economic load.

The nozzle governing is not very practical for multi-stage impulse turbines because in the high pressure stages, full admission is desirable to avoid partial admission losses. In addition, at the maximum loads the additional amount of steam required cannot be passed through the fist-stage, since the required additional number of nozzles are not available.

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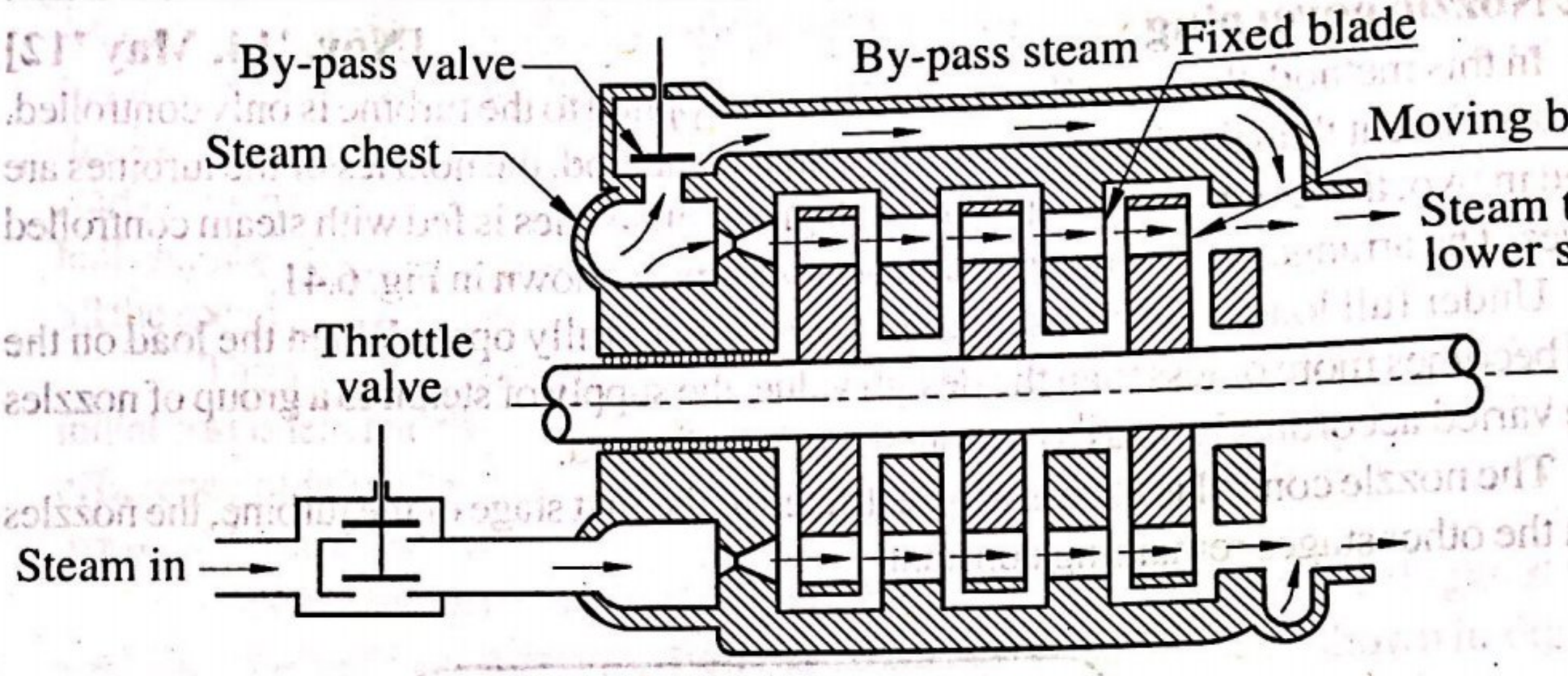


Fig. 6.42 By-pass governing

The arrangement of by-pass governing is shown in Fig. 6.42. The total amount of steam entering the turbine passes through the throttle valve which is under the control of speed governor. For all loads greater than the economic load, a by-pass valve is opened, allowing steam to pass from the first-stage nozzle box into the steam belt and so into the nozzles of the downstream stages. The by-pass valve is designed such that it is not opened until the lift of the throttle valve exceeds a certain value. The by-pass valve closes first as the load diminishes. The by-pass valve remains under the control of a speed governor for all loads within its range. More than one by-pass valve may be used.

Combined throttle governing and nozzle governing or throttle governing and by-pass governing are preferred in practice.

6.14.4 Comparison of throttle and nozzle governing [Sep. '13]

Throttle governing	Nozzle governing
1. Throttling losses are more.	1. Throttling losses are very negligible.
2. It is employed in both impulse and reaction turbine.	2. It is employed in impulse and also in reaction (if initial stage is impulse) turbine.
3. Partial admission losses are less.	3. Partial admission losses are more.
4. Heat drop available is less.	4. Heat drop available is more.
5. It is suitable for small medium and large turbine.	5. It is suitable for medium and large steam turbines if initial stage is impulse.

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6.15 Methods of attachment of blades to turbine rotor [May '15, May '13]

In design of steam turbine, the attachment of the turbine blades to the rotor is the most critical aspect because all the forces are transmitted through the attachment to the rotor. There are a number of methods for fixing turbine blades to the rotor. *The selected type of attachment must be able to resist the centrifugal and bending forces to which the blades may be subjected.* Blades which are loosely fastened to the rotor will amplify any vibrations induced in the blades, causing fatigue failure. The failure of one blade may lead to the destruction of the entire turbine. The various methods of blades attachment are given below.

(1) de Laval blade root attachment,
(2) Inverted - T attachment,
(3) Fir-tree or serrated blade root attachment,
(4) Straddle (single or Multiple fork) attachment,
(5) Side entry blades attachment,
(6) Shrouding strip attachment,
(7) Parsons integral blades.

The most of the above attachments are also used in gas turbine bladings.

(1) de Laval blade attachment :

The de Laval blade attachment is shown in Fig. 6.43. In this attachment, a number of holes are drilled in the rim of the wheel parallel to the axis. The slot for the blade tang is then finished by milling. The rectangular part connects the working section of blade to the rectangular tang. An integral shrouding provided at the tip of the blades to form continuous ring. This method of blades fastening is suitable for low pressure impulse steam turbines.

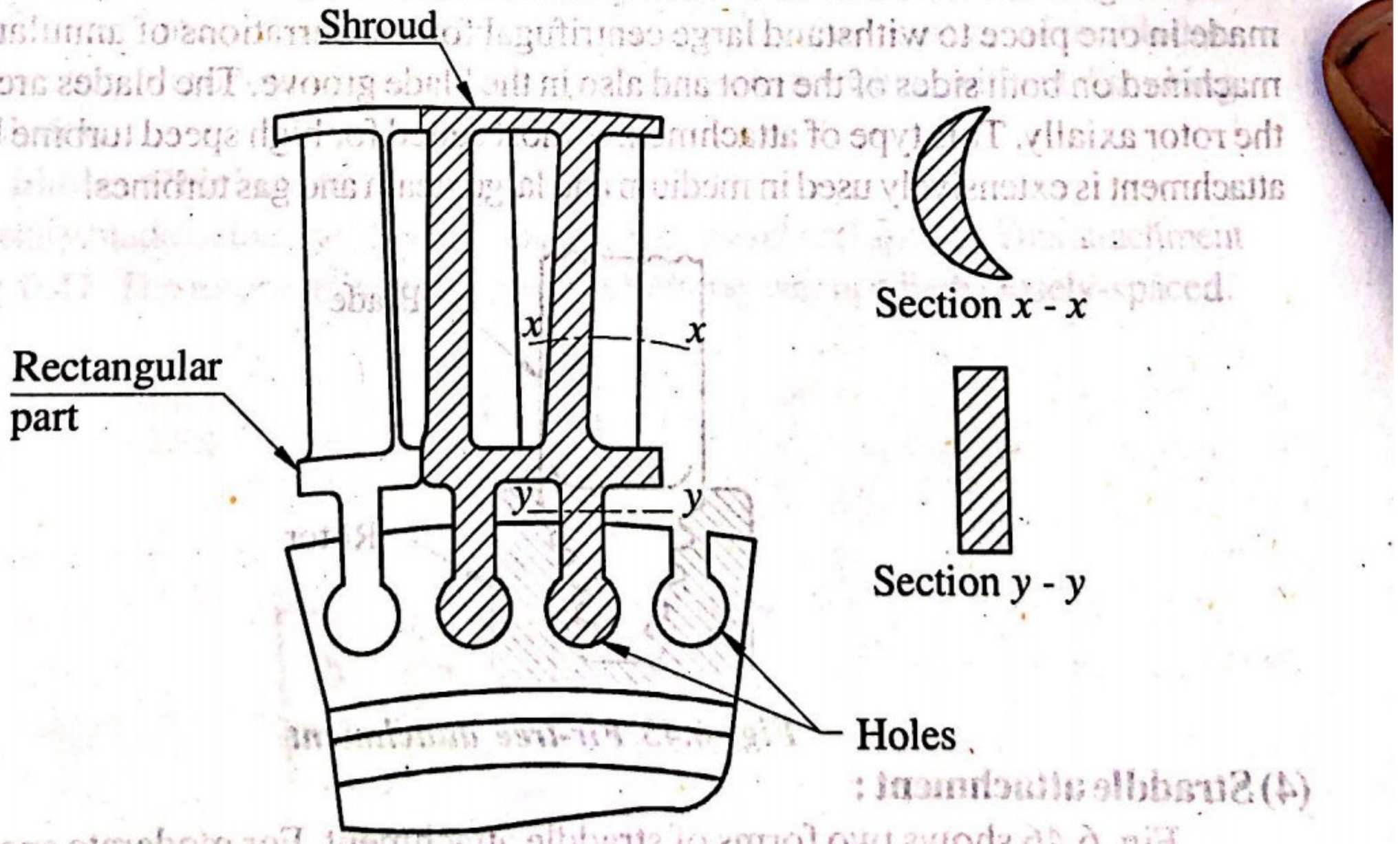


Fig. 6.43 de Laval blade attachment

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(2) Inverted-T attachment :

Fig. 6.44 shows inverted-T type and its various forms of blades attachment. The attachment shown in Fig. 6.44(a) is widely used for low pressure impulse turbines. Its modified form is shown in Fig. 6.44(b). It is used for the long low pressure blades of large steam turbines. This fastening is slipped radially into the slot provided on the disc or drum at a point widened for this purpose. The blades are then pushed along in the slot until the last blade is inserted, whereupon a special stop piece is put in place.

An inverted T-shaped attachment used in the high pressure impulse stages of large turbines is shown in Fig. 6.44(c).

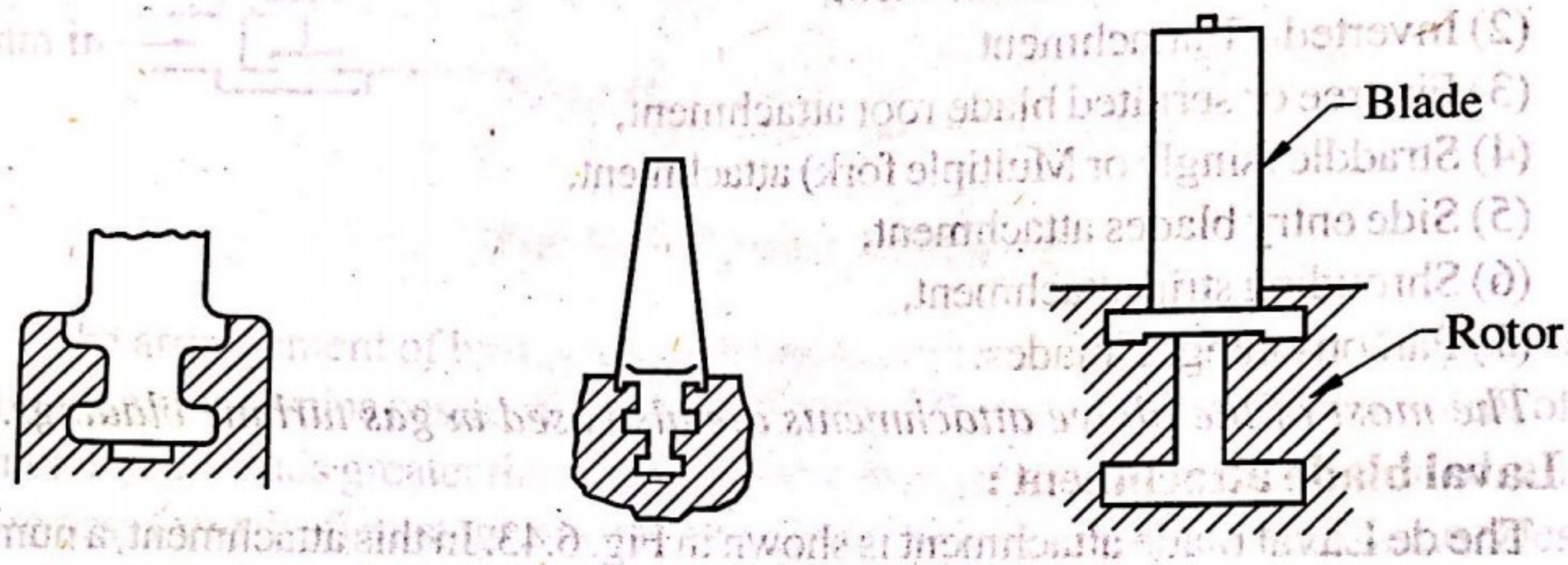


Fig. 6.44 Inverted-T attachment

(3) Fir-tree or serrated blade root attachment :

Fig. 6.45 shows a fir-tree or serrated blade root attachment. The blade and spacer is made in one piece to withstand large centrifugal forces. Serrations of annular fir-tree are machined on both sides of the root and also in the blade groove. The blades are inserted on the rotor axially. This type of attachment is most suited for high speed turbine blades. This attachment is extensively used in medium and large steam and gas turbines.

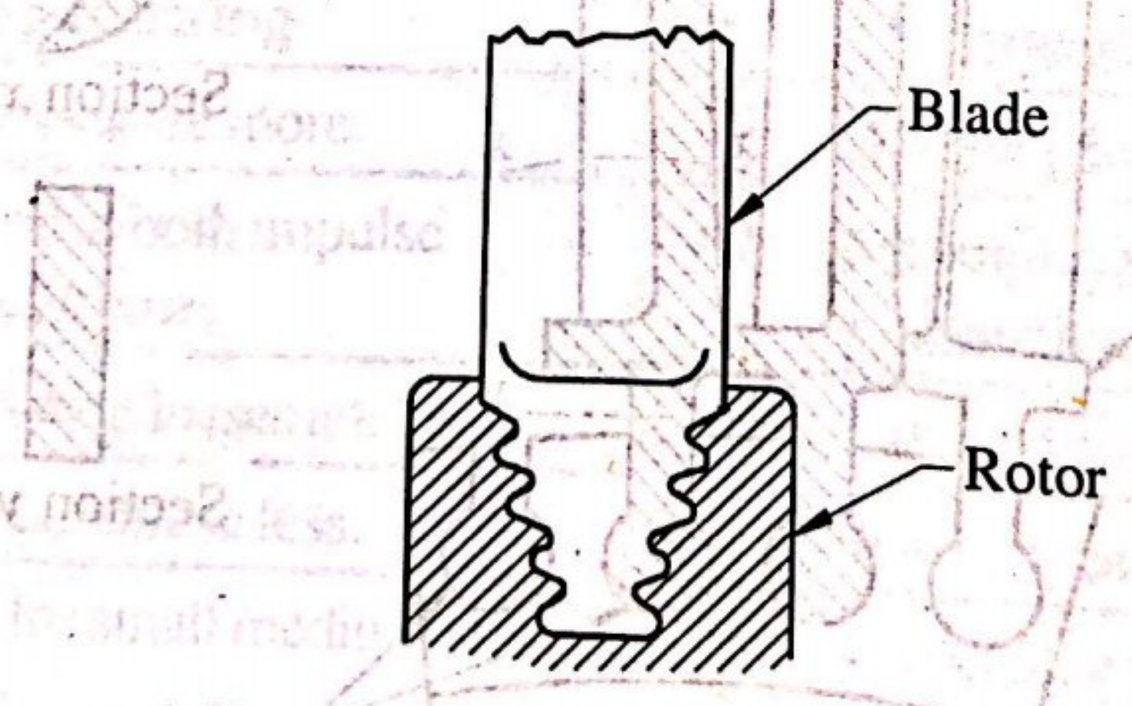


Fig. 6.45 Fir-tree attachment

(4) Straddle attachment :

Fig. 6.46 shows two forms of straddle attachment. For moderate speeds and short blades the type shown in Fig. 6.46(a) is used. The blade and spacer must be machined from

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one piece of metal. The blade is forked and fits tightly on to a projecting tongue on the wheel. The rivets are slightly countersunk and staggered.

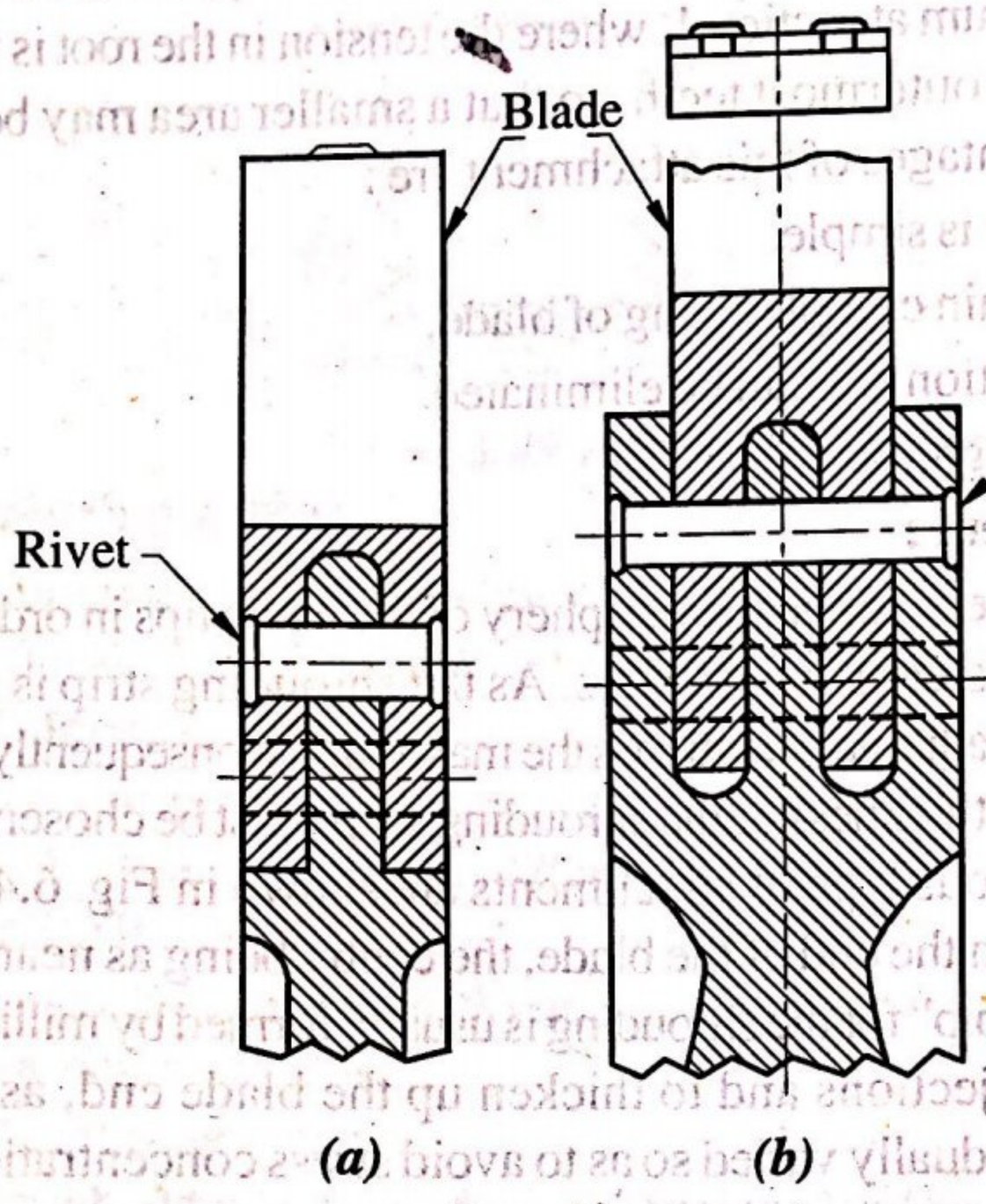


Fig. 6.46 Straddle blade attachments

When the shear stress in the rivets is excessive, as with higher speeds or longer blades, the attachment shown in Fig. 6.46(b) (multiple fork) is used, the rivets then being in quadruple shear. In straddle attachment, there is no need to cut any gap for the insertion of the blades. Moreover, if necessary one or more blades may be removed from the wheel without disturbing the remaining blades.

(5) Side entry blades attachment :

A side entry blades attachment is used for very high peripheral speeds. This attachment is shown in Fig. 6.47. The rotor carries a comparatively heavy rim in which closely-spaced

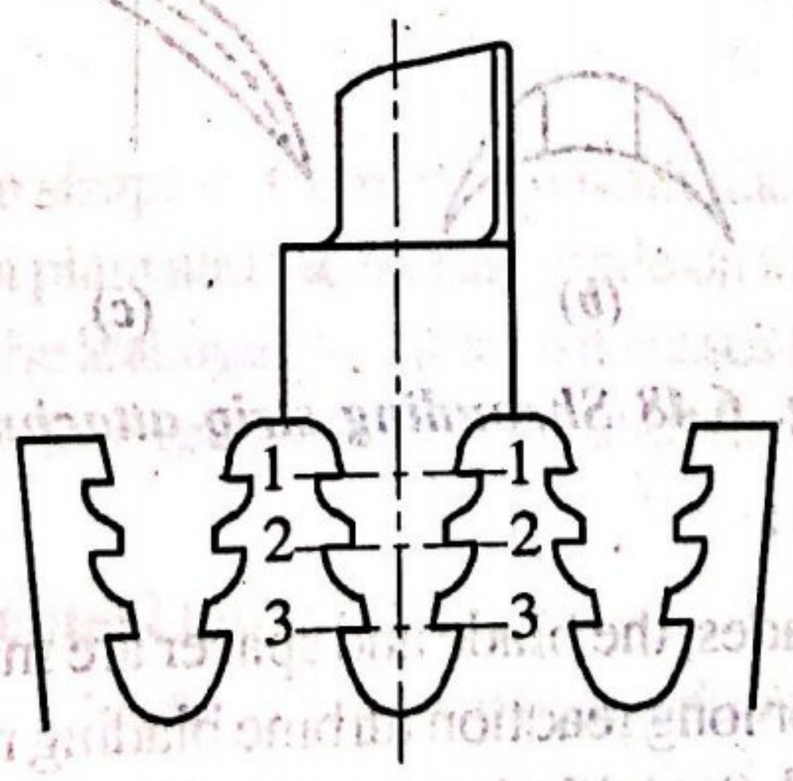


Fig. 6.47 Side entry blade attachment

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slots are cut, one to each blade. The slots may be either parallel or skew to the axis, depending on the shape of the blade root profile. Considering the blade only, the area of the material under tension is made a maximum at section 1, where the tension in the root is the maximum. Part of the pull is taken by the outermost teeth, so that a smaller area may be employed at section 2 and so on. The advantages of this attachment are :

(1) Blades replacement is simple,

(2) It is possible to obtain complete ring of blade,

(3) Bending across section 1-1 can be eliminated.

The cost of this attachment is high.

(6) Shrouding strip attachment :

A shroud is a band placed around the periphery of the blade tips in order to stiffen the blades and prevent spillage of steam over the tips. As the shrouding strip is attached to the blades at the extreme radius, the radial acceleration is the maximum. Consequently, the centrifugal force is often very large and the thickness of the shrouding strip must be chosen with reference to the bending stress. The various type of attachments are shown in Fig. 6.48. Fig. 6.48(a) shows a round pin machined on the end of the blade, the centre being as near to the centroid as possible. In Fig. 6.48(b) the "pip" for the shrouding is usually formed by milling. Sometimes, it is necessary to use two projections and to thicken up the blade end, as shown in Fig. 6.48(c). The section is then gradually varied so as to avoid stress concentration.

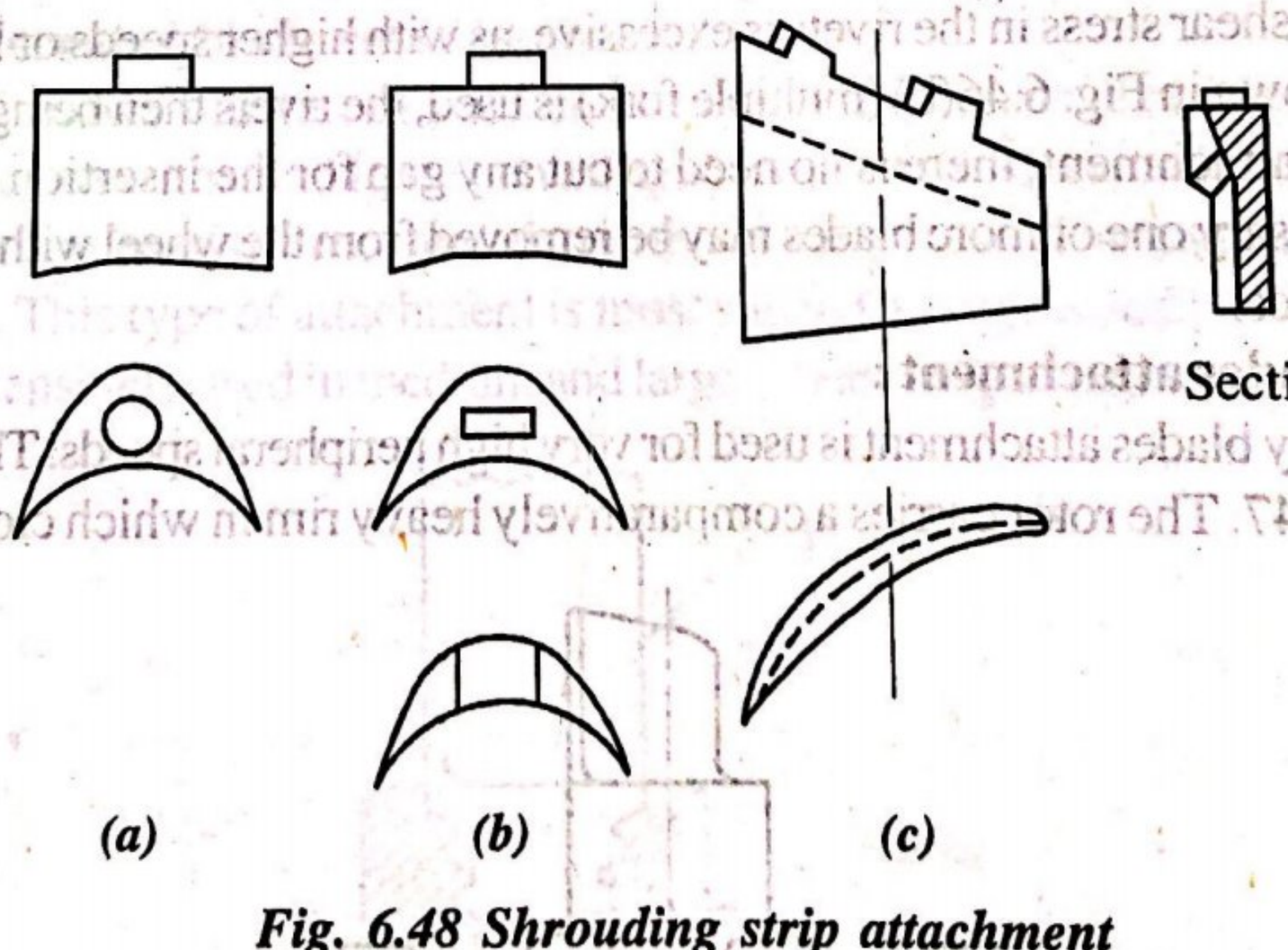


Fig. 6.48 Shrouding strip attachment

(7) Parsons integral blades :

In Parsons integral blades, the blade and spacer are made from one piece as shown in Fig. 6.49. It is more suitable for long reaction turbine blading running at high speeds. The root of the blade is lozenge shaped. Such blades are rolled by a discontinuous process and made from a rectangular bar. The blade is pickled, polished and the serrations machined on the root.

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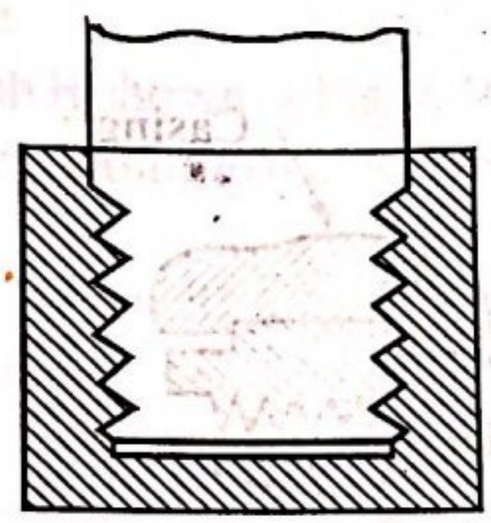


Fig. 6.49 Parsons integral blades

[May '15]

6.16 Labyrinth packing

A small radial clearance space must be provided between rotating and stationary components (i.e. shaft and casing) to separate them. Steam or air will leak through this clearance if pressure difference exists across them. *Glands and seals are the devices used in turbines to prevent or reduce the leakage of steam or air between rotating and stationary components that have a pressure difference across them.* The various types of glands and seals are listed below.

(1) Labyrinth gland or packing,
(2) Carbon ring gland,
(3) Water sealed gland.

Labyrinth gland is better than carbon ring gland because it can withstand higher steam temperature. The labyrinth gland consists of a ring with a series of machined fins that form a number of fine annular restrictions which cuts on the throttling principle. The various forms of labyrinth gland are shown in Fig. 6.50.

(1) Axial clearance gland :

Fig. 6.50(a) shows the axial clearance or facial type of labyrinth packing. In this type, the rings of non-ferrous metal are caulked into the gland casing. Shallow collars about 0.31 cm wide are turned on the shaft and the leakage is controlled by the axial clearance C_a. This type of packing is used at the end of the turbine (usually the H.P. end) adjacent to the adjusting block.

(2) Plain labyrinth :

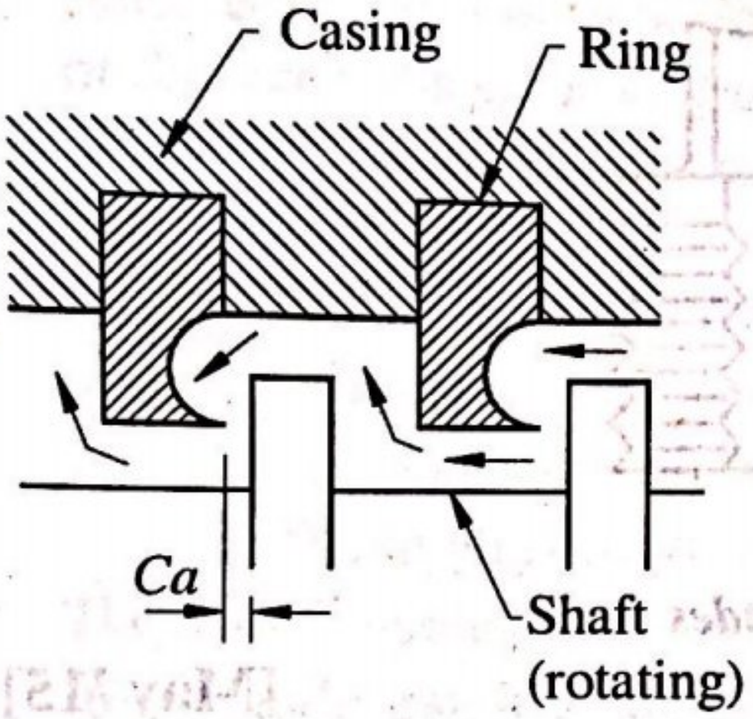
Plain labyrinth is the simplest form of labyrinth packing as shown in Fig. 6.50(b). In this labyrinth, the shaft is kept plain and the fins are made on the ring. The number of restrictions and radial clearance control the leakage. As the steam passes through the restrictions, throttling process occurs and it results into drop of pressure and in a reduction in leakage of steam or air.

(3) Stepped and double stepped labyrinth :

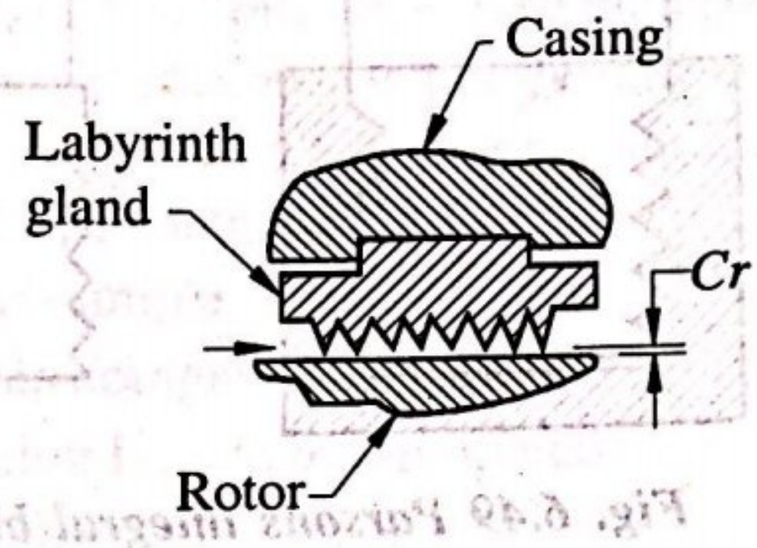
Fig. 6.50 (c) and Fig. 6.50(d) show stepped and double stepped labyrinth respectively. Stepped labyrinth gland can only be employed where axial differential expansion between the rotor and casing are small while the double stepped labyrinth gland can accept a large axial differential expansion. Two of the three fins per pitch is sufficient for effective restriction.

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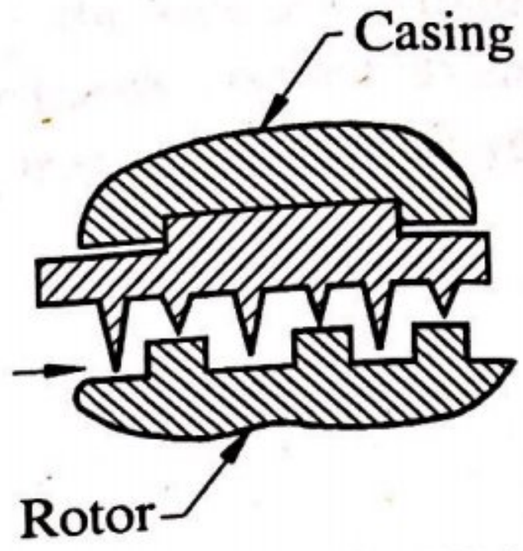
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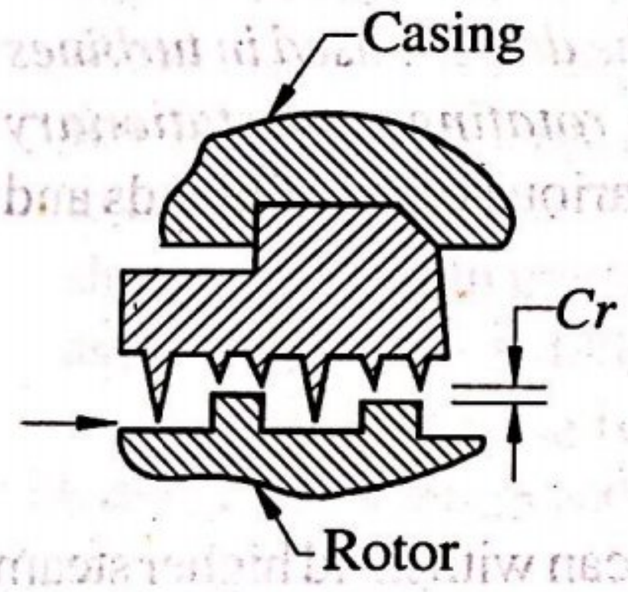
(a) Axial clearance gland



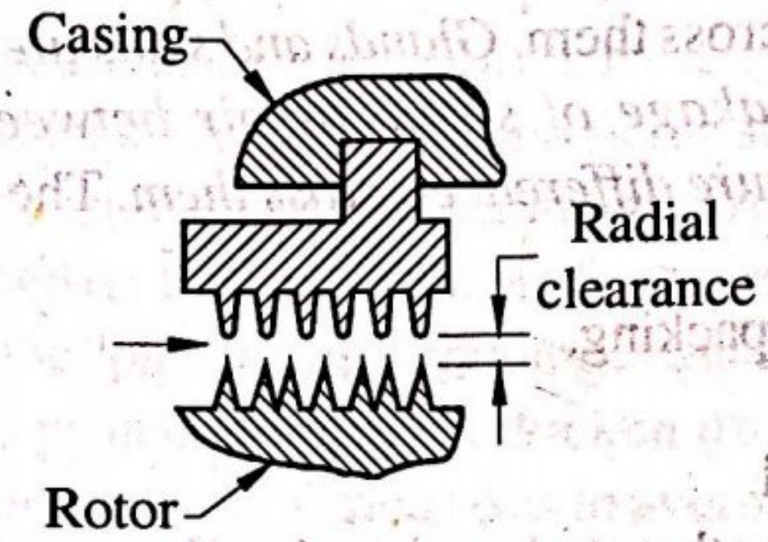
(b) Plain labyrinth



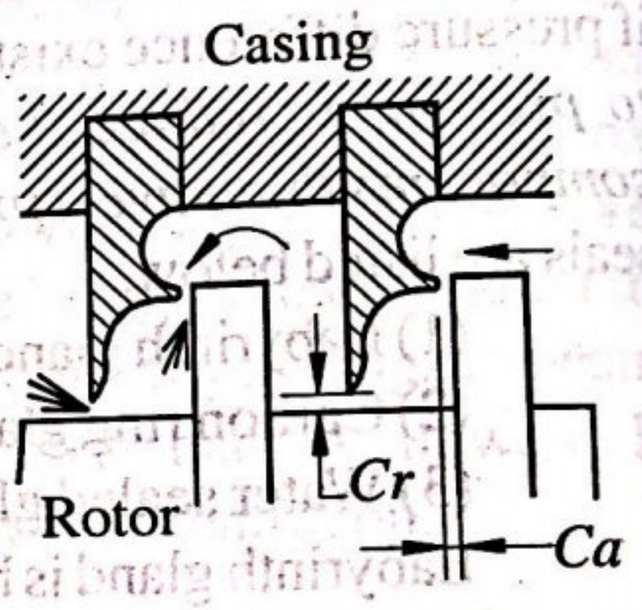
(c) Stepped labyrinth



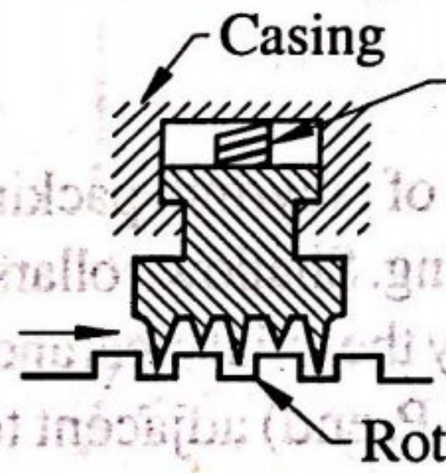
(d) Double stepped labyrinth



(e) Vernier labyrinth



(f) Radial & axial



(g) Spring back



(h) Staggered

Fig. 6.50 Forms of labyrinth glands

(4) Vernier labyrinth :

Fig. 6.50(e) shows vernier labyrinth in which the seal ring and the shaft are finned. This type of packing is not as efficient as the stepped seal but it is independent of differential expansion.

(5) Radial and axial labyrinth :

Fig. 6.50(f) shows this type of packing. It is the combination of radial and axial packing which gives two restrictions per ring. It is used only at the end adjacent to the adjusting block where the axial clearance is under control.

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(6) Spring back labyrinth :

Spring back labyrinth is shown in Fig. 6.50(g). In this type, the gland rings are spring loaded in order to reduce the rubbing effect with close clearance glands.

(7) Staggered labyrinth :

Fig. 6.50(h) shows staggered type labyrinth gland. Generally, it is used to insure that no kinetic energy is carried over from the preceding restrictions.

The flow through a labyrinth gland is a function of the initial pressure and temperature, the final pressure, and the clearance area under the restrictions.

6.17 Losses in steam turbines [Nov. '14, Sep. '13, May '12]

In actual practice, the work done by a turbine is less than isentropic heat drop of the steam used. There are several factors, which reduce the output of the turbine, are known as ***internal losses***. In addition to this, mechanical losses and leakage of steam through end glands further reduce the useful workdone on the turbine coupling.

The internal losses in steam turbine may be listed as follows :

(1) Regulating valve losses
(2) Nozzle or fixed blade losses
(3) Moving blade losses
(4) Disc friction losses
(5) Partial admission or blade windage losses
(6) Clearance losses
(7) Carry-over losses
(8) Residual or leaving velocity losses
(9) Moisture loss
(10) Radiation and conduction losses

The external losses in steam turbine may be listed as follows :

(1) Mechanical losses, and (2) Leakage losses from the end seals.

6.17.1 Internal losses

(1) Regulating valve losses :

Steam from boiler, first passes through the main stop valve and the regulating valves before it enters to the turbine. The flow through these being accompanied by pressure losses. Steam gets throttled adiabatically with constant enthalpy. Fig. 6.51 shows the expansion of

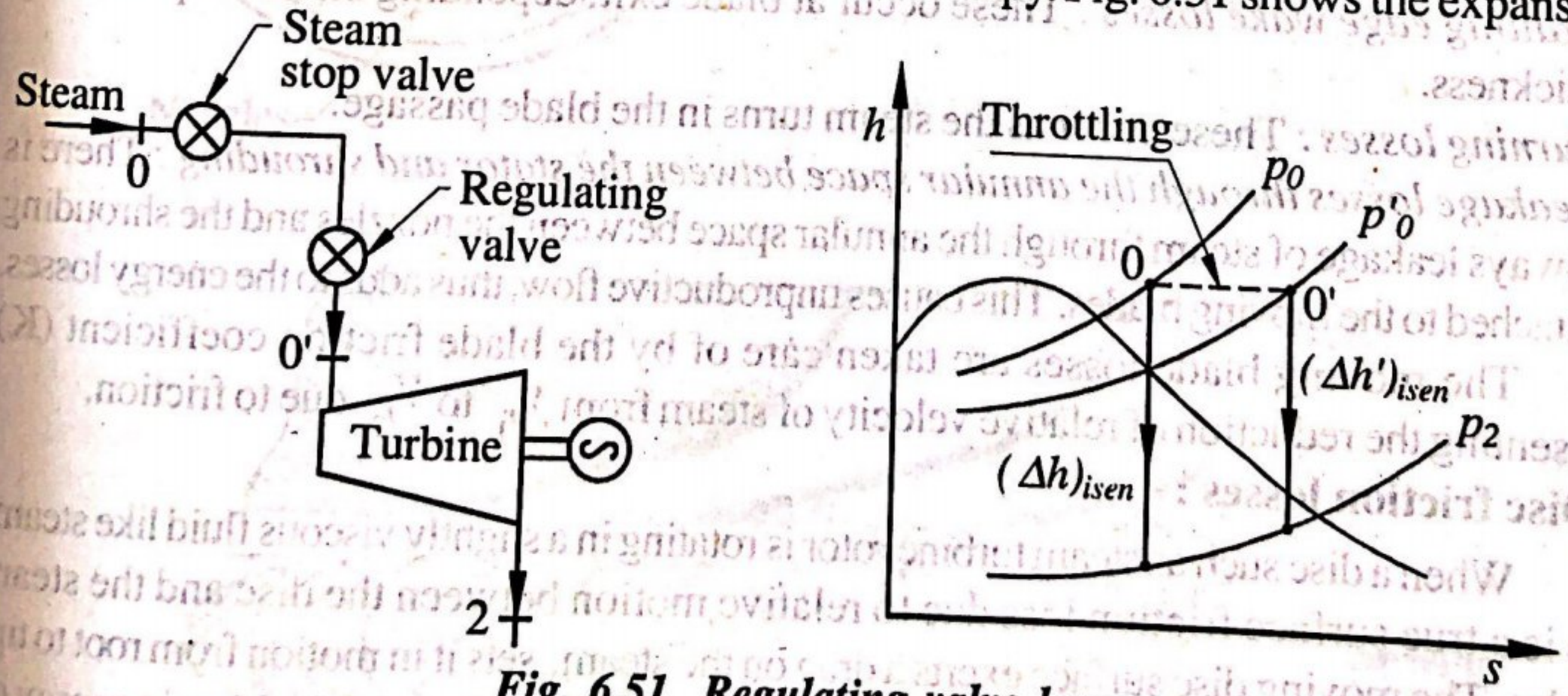


Fig. 6.51 Regulating valve losses

steam through regulating valves on *h-s* diagram from initial pressure p_0 to p'_0. Due to throttling in the regulating valves, the isentropic heat drop decreases from $(\Delta h)_{isen}$ to $(\Delta h')_{isen}$ as

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shown in Fig. 6.51. Hence loss in available energy equal to

$$(\Delta h_{isen})_{valve} = (\Delta h)_{isen} - (\Delta h')_{isen}$$

The pressure drop varies from 3 to 5% of the inlet steam pressure p_0.

(2) Nozzles or Fixed blade losses :

These losses occur when the steam flows through the nozzle or fixed blade of turbine. These losses take place due to friction in the nozzle and the formation of eddies. The friction losses in nozzles were mentioned earlier in chapter of nozzles.

Losses in nozzles produce heat, hence the heat content of steam at the nozzle outlet will be more than the theoretical value. Therefore, the actual heat drop in the nozzle will be less than theoretical value resulting reduction in velocity at nozzle outlet.

Let V_{1t} = theoretical velocity at nozzle exit

V_{1a} = actual velocity at nozzle exit.

$$\therefore \text{Losses of energy in nozzle} = \frac{V_{1t}^2 - V_{1a}^2}{2 \times 1000} \text{ kJ/kg}$$

The nozzle loss depends on its size, surface roughness, its length, roundness of entrance, divergence angle, space between nozzle, moisture and trailing edge.

(3) Moving blade losses :

Moving blade losses are caused by the following factors.

(a) *Blade frictional losses :* Steam flow experiences a frictional resistance when it passes through the blade passage. This loss depends upon the blade chord and the roughness of the blade surface.

(b) *Impingement losses :* If the entry of steam to the leading edges of the blades is not smooth, the eddies will be formed which causes loss of energy.

(c) *Trailing edge wake losses :* These occur at blade exit, depending on its shape and tip thickness.

(d) *Turning losses :* These occur as the steam turns in the blade passage.

(e) *Leakage losses through the annular space between the stator and shrouding :* There is always leakage of steam through the annular space between the nozzles and the shrouding attached to the moving blades. This causes unproductive flow, thus adds to the energy losses.

The moving blade losses are taken care of by the blade friction coefficient (K) representing the reduction of relative velocity of steam from V_{r_1} to V_{r_2} due to friction.

(4) Disc friction losses :

When a disc such as steam turbine rotor is rotating in a slightly viscous fluid like steam, there is a true surface friction loss due to relative motion between the disc and the steam particles. The moving disc surface exerts a drag on the steam, sets it in motion from root to tip, and produces a definite circulation as shown in Fig. 6.52. Some part of the kinetic energy of steam is lost due to this friction.

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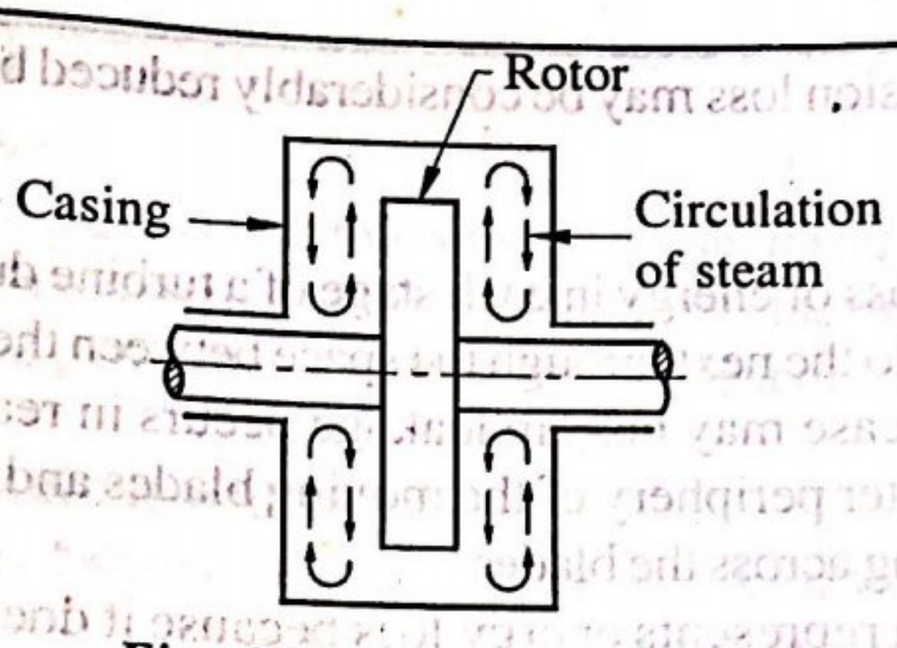


Fig. 6.52 Disc friction losses

(5) Partial admission or Blade windage losses :

In the high pressure stage of the steam turbine, a relatively small flow area is required. In such a turbine nozzles are provided only over a part of the blade periphery. Due to this some blade channels are partially filled with steam. This results in partial admission and some of the blades which are not under the influence of steam jets and idly churn the steam. This gives rise to the formation of eddies in those channels as shown in Fig. 6.53 causing a loss to the turbine. This is termed as blade windage loss.

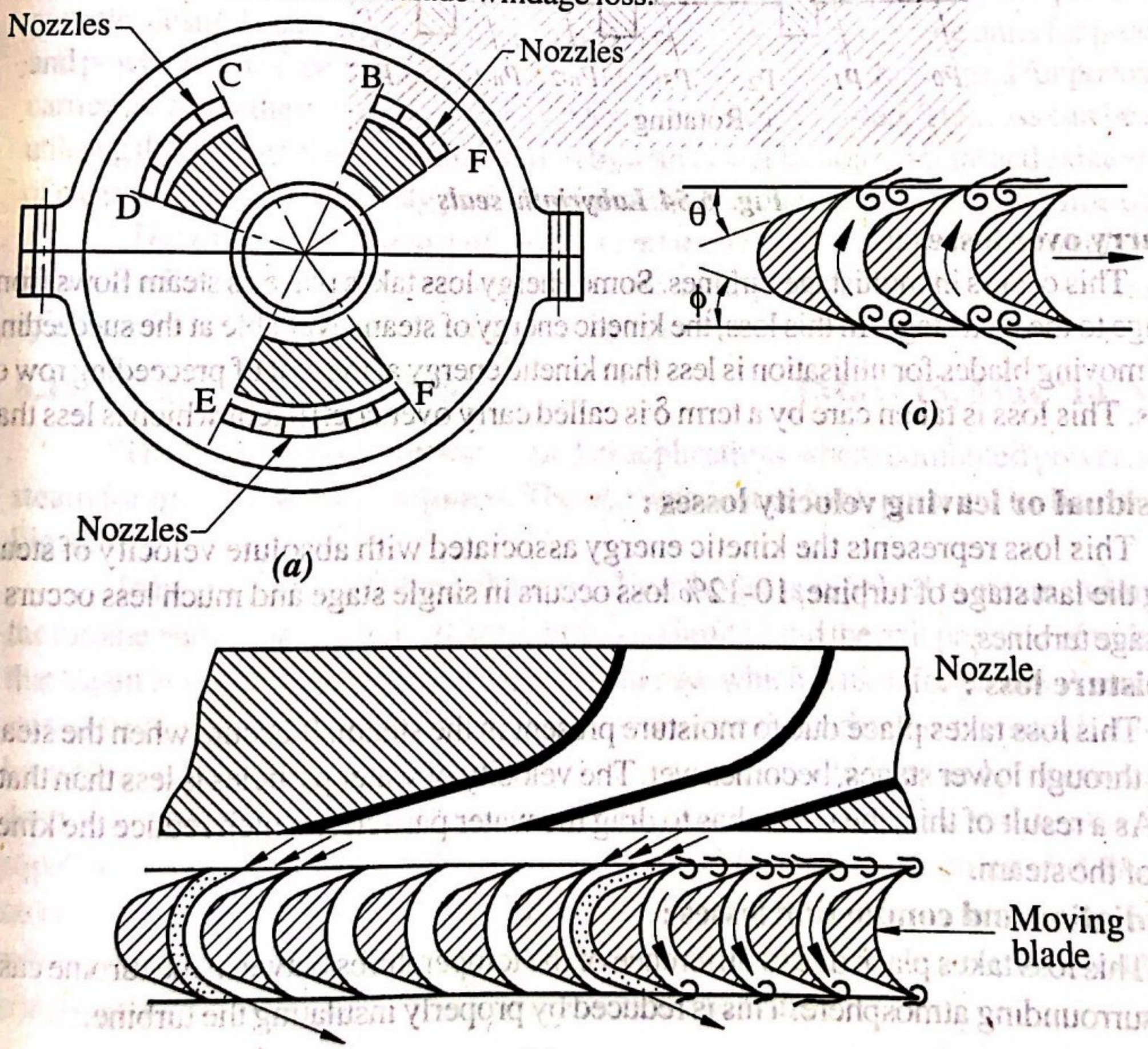


Fig. 6.53 Partial admission loss

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The partial admission loss may be considerably reduced by enclosing the blades on both sides.

(6) Clearance losses :

There is a small loss of energy in each stage of a turbine due to the leakage of steam from one wheel chamber to the next through the space between the diaphragm and the shaft, or the wheel hub, as the case may be. Tip leakage occurs in reaction stages through the clearance between the outer periphery of the moving blades and the casing because of the pressure difference existing across the blades.

The leaked steam represents energy loss because it does not work on the blades. Both diaphragm and tip leakage can be minimized by reducing the radial clearances, but it must avoid rubbing or metal-to-metal contact. To reduce leakage through diaphragm, special labyrinth packing are employed as shown in Fig. 6.54.

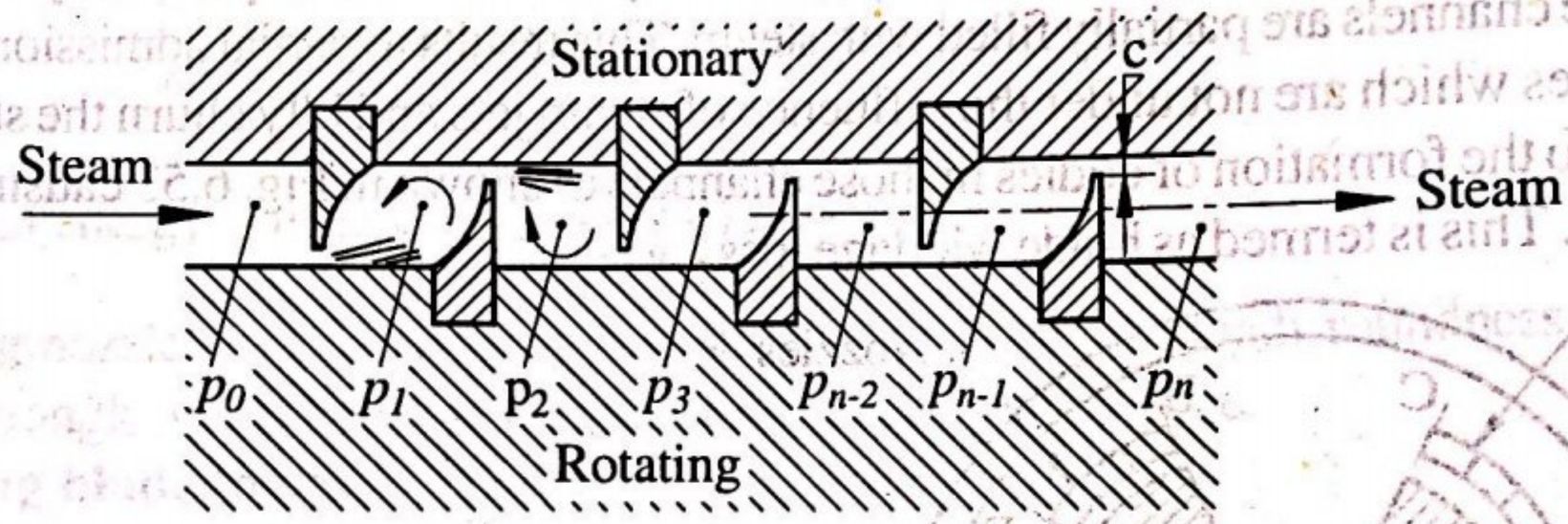


Fig. 6.54 Labyrinth seals

(7) Carry over losses :

This occurs in multistage turbines. Some energy loss takes place as steam flows from one stage to the next stage. In this loss, the kinetic energy of steam available at the succeeding row of moving blades for utilisation is less than kinetic energy at the exit of preceeding row of nozzles. This loss is taken care by a term δ is called carry over coefficient which is less than unity.

(8) Residual or leaving velocity losses :

This loss represents the kinetic energy associated with absolute velocity of steam leaving the last stage of turbine. 10-12% loss occurs in single stage and much less occurs in multi-stage turbines.

(9) Moisture loss :

This loss takes place due to moisture present in the steam. It occurs when the steam, passing through lower stages, becomes wet. The velocity of water particles is less than that of steam. As a result of this, the steam has to drag the water particles, which reduce the kinetic energy of the steam.

(10) Radiation and conduction losses :

This loss takes place due to difference of the temperatures between the turbine casing and the surrounding atmosphere. This is reduced by properly insulating the turbine.

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6.17.2 External losses

(1) Mechanical losses :

These losses occur due to friction between the shaft and wheel bearing. These losses also include energy loss in governing mechanisms and any other machine coupled to the turbine shaft such as the main oil pump, etc. These losses can be reduced by lubricating the moving parts of the turbine.

(2) Leakage losses from the end seals :

There is leakage of steam from two ends of the turbine where the turbine shaft projects out from the turbine casing. This leakage is due to pressure difference between inside the casing and the outside atmosphere. There is steam leakage through the end of labyrinth seals.

6.18 Special types of steam turbines

In several industries such as textile, chemical, paper and pulp, jute mills, sugar factories and so on in which there is a dual demand of power and steam for process heating and drying. For this type of work, previously, the steam was generated at a moderate pressure for power purposes and saturated steam for process work was generated separately at a pressure which gave the desired heating temperature. There were having two separate units for process heat and power. In this type of system much of the heat of the steam that is used for power work is carried away by the cooling water in condenser and that is wasted. This loss can be saved by utilising the exhaust steam in process work by a suitable selection of initial and exhaust pressure of steam and thus by this way power and process work are achieved at the same time.

The thermal efficiency of such a combined power and heating plant is very high because all or part of the steam is condensed in the process work and no or less amount of steam is condensed in a separate condenser.

6.19 Back pressure turbines [May '15, May '14, May '13]

The back pressure turbine is used in applications where combined power and heat in steam for process work is required. The arrangement of back pressure turbine is shown in Fig. 6.55.

In back pressure turbine, the steam from boiler is supplied to steam turbine through the turbine valve. The expansion in this turbine is limited and the exit pressure of turbine is such that steam leaving the turbine is in superheated state which is used for process heating for high rate of heat transfer. The superheated steam leaving the turbine is not suitable for process heating, a desuperheater is introduced in between the exit of turbine and process heater. In this desuperheater, a jet of water, thermostatically controlled, is sprayed and due to contact with superheated steam spray water is evaporated and the steam is cooled and thus becomes saturated steam. Saturated steam from the desuperheater enters the process heater and is entirely condensed by giving heat to the heating surface. Depending upon the conditions, the condensed steam may or may not be returned to the boiler.

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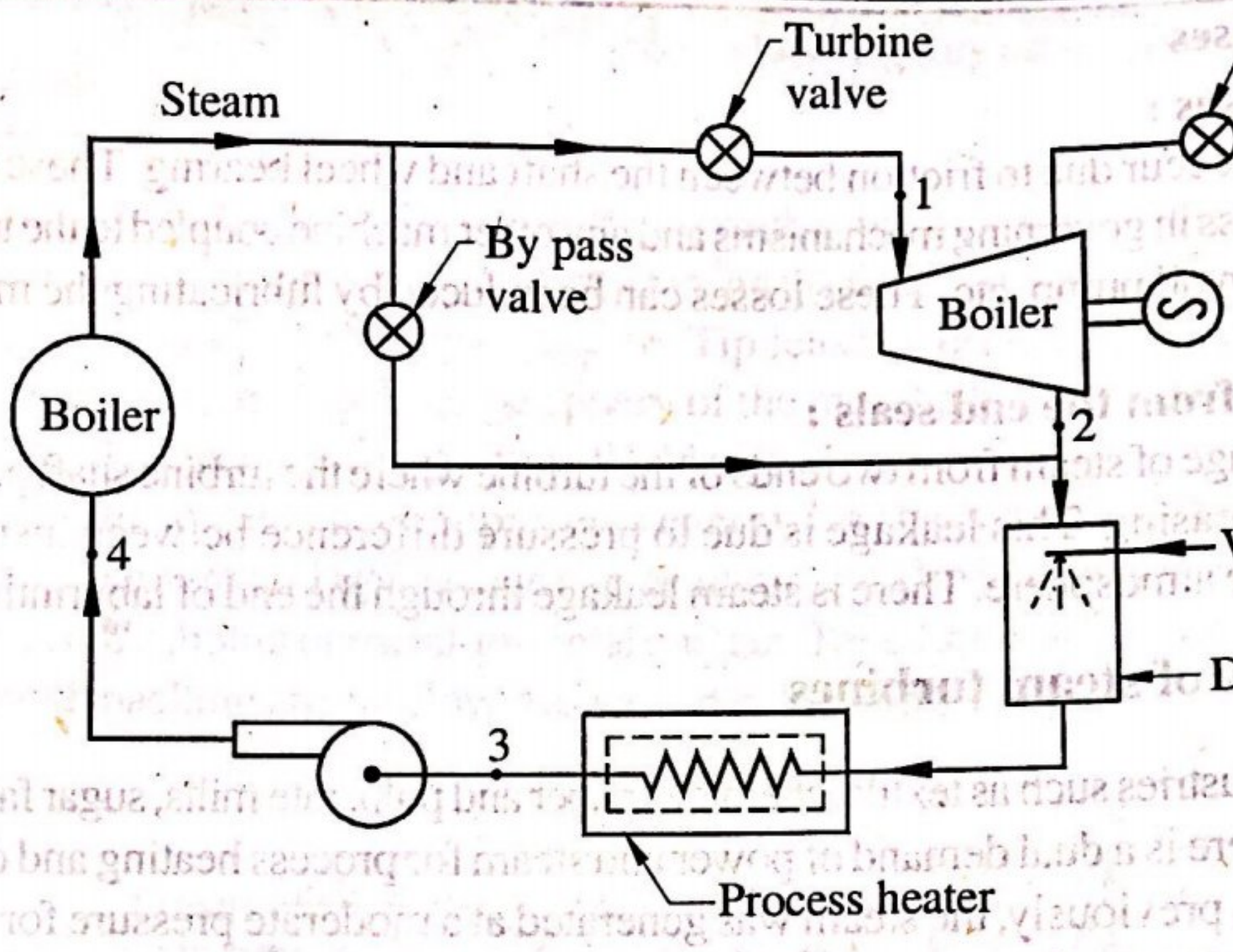
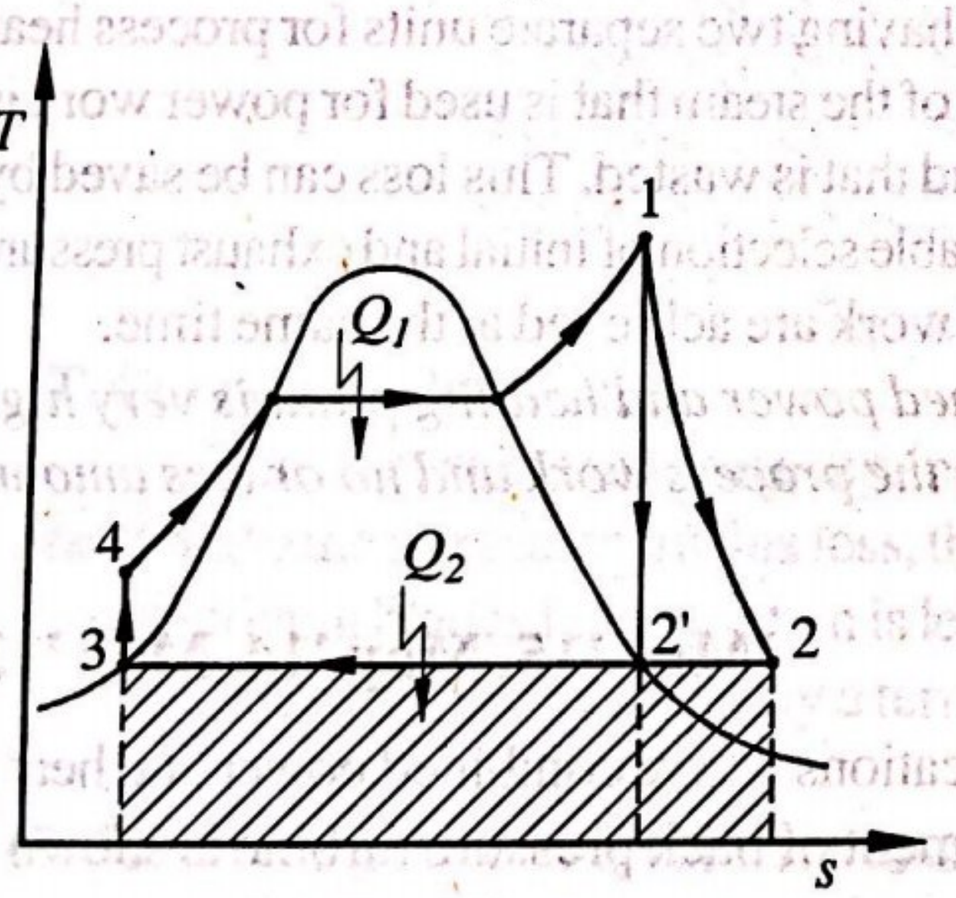
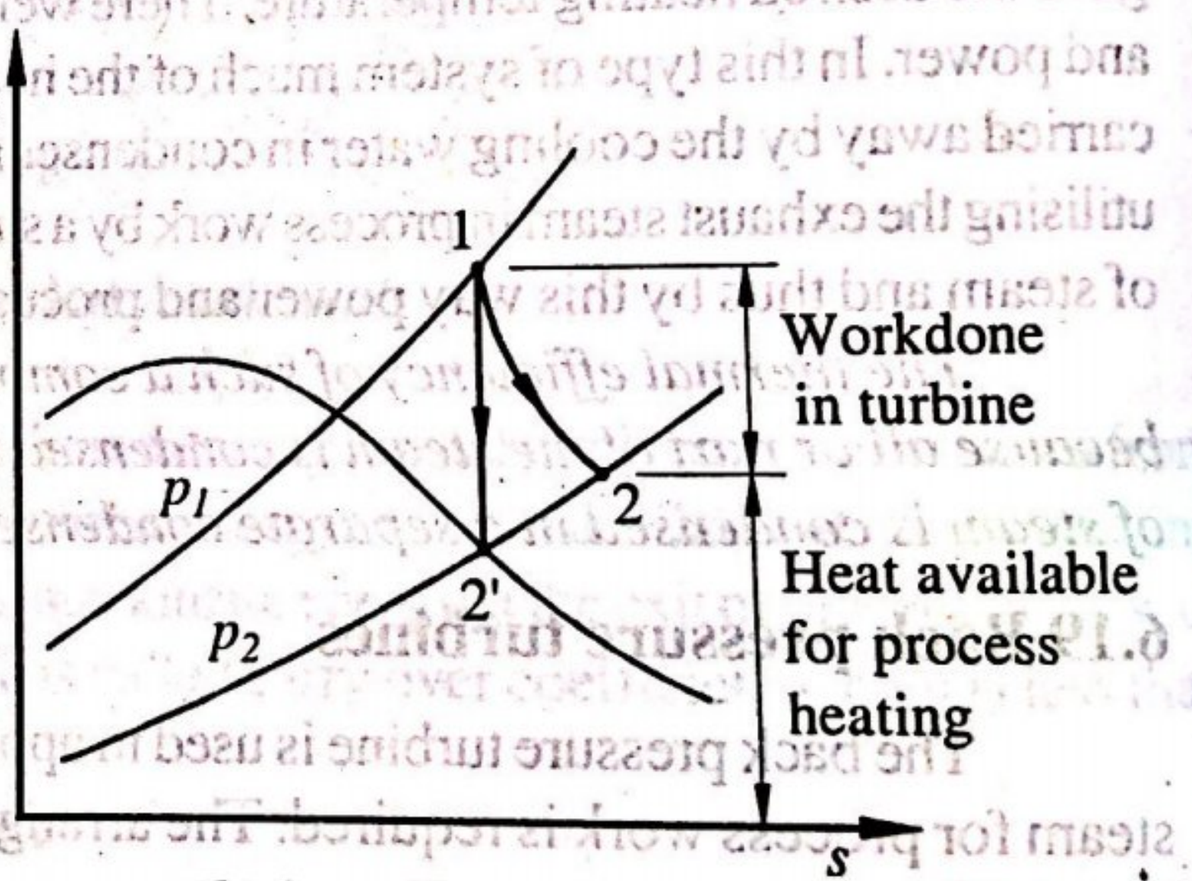


Fig. 6.55 Arrangement of back pressure turbine



(a) T - s diagram



(b) h - s diagram

Fig. 6.56 Representation of back pressure turbine

In this arrangement, generally, the steam required for power generation will not always be equal to that required for process work. To avoid variation in exhaust pressure and therefore, of the steam saturation temperature, some methods of controlling the exhaust steam pressure must be employed. If the available exhaust steam is too small, live steam may be passed through the by pass valve (pressure reducing valve) into the desuprheater. If the quantity of exhaust steam is in excess of what is strictly necessary for the process work, then the excess steam may be blown into a feed tank or to atmosphere by regulating the exhaust valve. If there is no power requirement by the industry, the by-pass valve by pass the steam directly from the boiler to desuperheater for process heating.

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Fig. 6.56 shows the *T-s* and *h-s* diagrams for such a cycle. Let Q_1 is the rate of heat energy supplied to steam boiler, and Q_2 is heat energy utilized for process heating.

$$\therefore Q_1 = m_1 (h_1 - h_4)$$

where m_1 = rate of steam generation in boiler, *kg/s*

Turbine work, $W_T = m_2 (h_1 - h_2)$

where m_2 = rate of steam supplied to turbine, *kg/s*

If no steam is by passed, then $m_1 = m_2$

Heat energy utilized for process heating,

$$Q_2 = m_1 (h_2 - h_3)$$

Neglecting the pump work,

Efficiency of the plant, $\eta = \dfrac{W_T + Q_2}{Q_1}$

$$\eta = \frac{m_2 (h_1 - h_2) + m_1 (h_2 - h_3)}{m_1 (h_1 - h_4)} \quad ...(6.28)$$

The back pressure turbines are quite small with respect to their power output because they exhaust pressure is high and specific volume of steam is less. Hence they are cheap in terms of cost per MW compared to condensing turbines of the same power.

6.20 Pass out turbine or Extraction turbine [Sep. '13, May '12]

In many cases, the power available from the back pressure turbine is considerably less than that required in a factory. This may be due to relatively high back pressure, or small heating requirement or both. Pass-out turbines are used in these cases, where a certain quantity of steam is continuously extracted from the turbine at an intermediate stage for heating purposes at the desired temperature and pressure.

The arrangement of this turbine is shown in Fig. 6.57. In this system whole of steam from boiler enters into the High Pressure (H.P.) turbine and the steam is allowed to expand upto the pressure requirements for process heating. A part of the steam from exhaust of H.P. turbine is extracted for process heating and remainder is supplied to Low Pressure (L.P.) turbine through the by pass valve which regulates the steam supply. This steam is allowed to expand in low pressure turbine upto the condenser pressure for power generation. The condensate from condenser is returned to boiler by feed pump-1. A pressure regulator is introduced into the system which regulates and maintains the pressure of steam as per the pressure requirement of process heat. To facilitate the operation of pass-out turbine for variable extraction of steam requirements either the nozzle governing or throttle governing is used.

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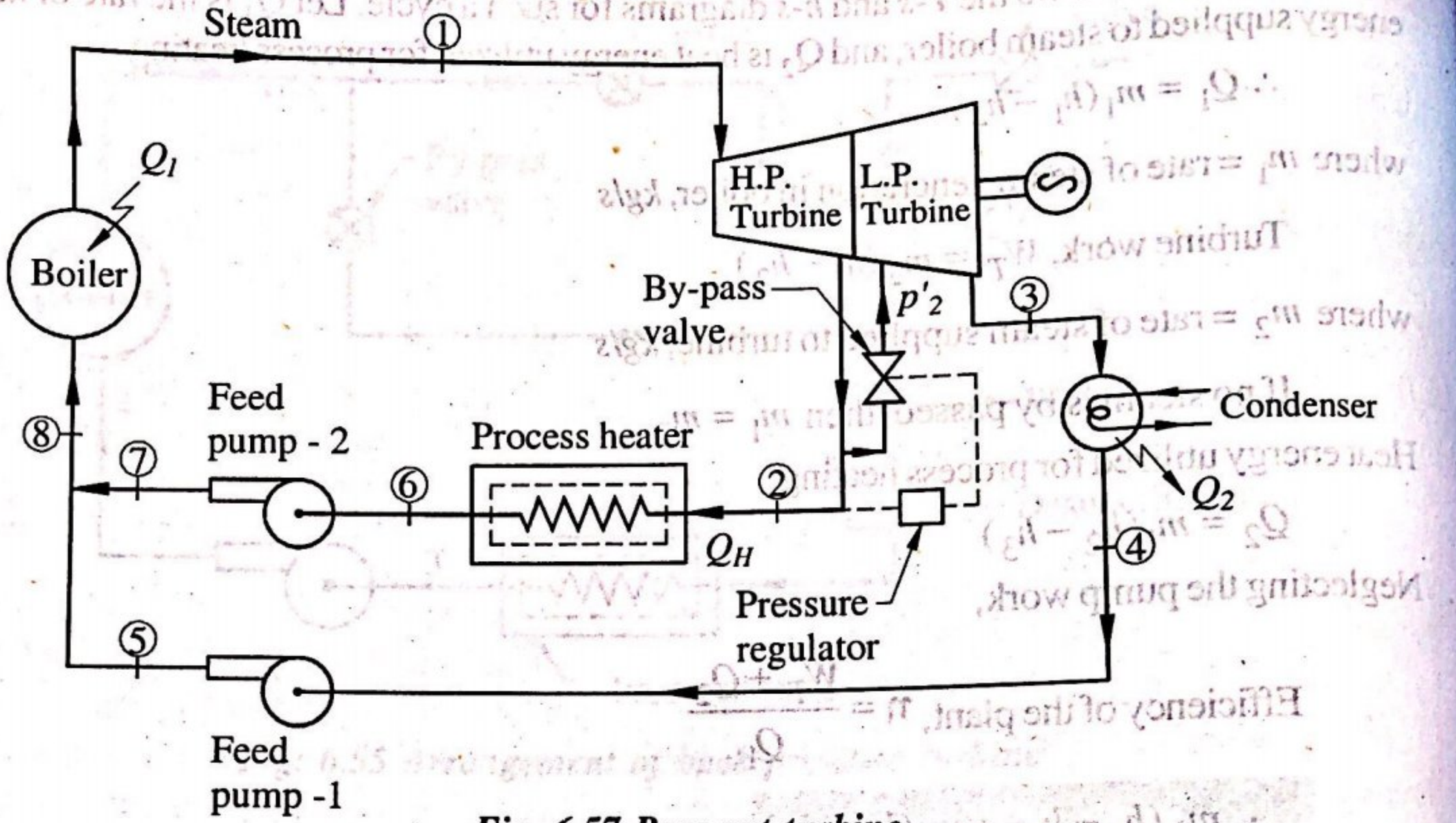
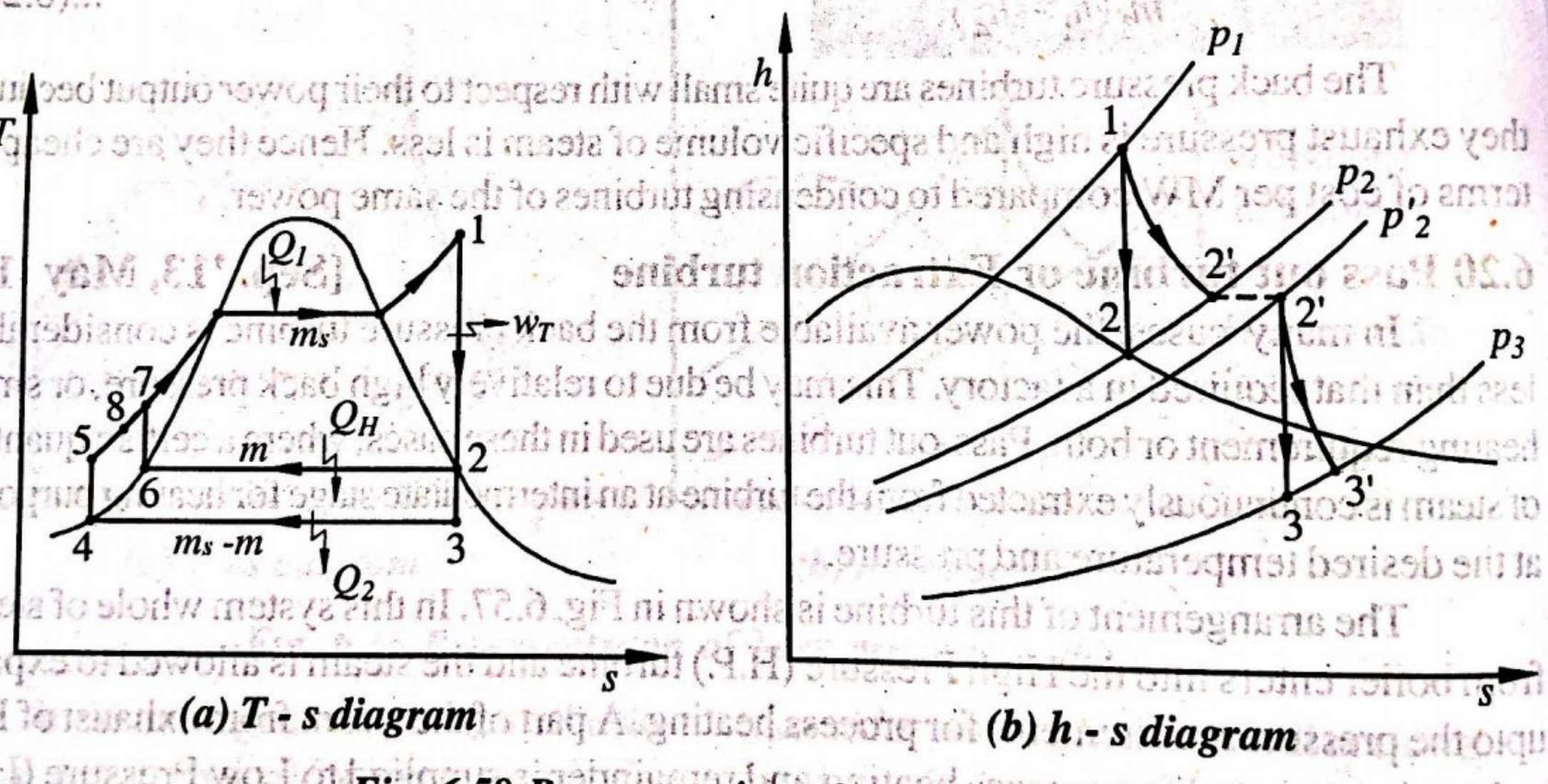


Fig. 6.57 Pass-out turbine



(a) T- s diagram *(b) h.- s diagram*

Fig. 6.58 Representation of pass out turbine

The process of pass out turbine is represented on *T-s* and *h-s* diagram as shown in Fig. 6.58.

m_s = mass of steam generated from boiler, *kg/s*

m = mass of steam extracted for process heating, *kg/s*

Heat energy supplied to the boiler,

$$Q_1 = m_s (h_1 - h_8)$$

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Heat energy rejected to condenser,

$$Q_2 = (m_s - m)(h_3 - h_4)$$

Heat energy utilized for process heating,

$$Q_H = m(h_2 - h_6)$$

Turbine work, $W_T = m_s(h_1 - h_2) + (m_s - m)(h_2 - h_3)$

Pump work, $W_p = (m_s - m)(h_5 - h_4) + m(h_7 - h_6)$

Efficiency of the plant,

$$\eta = \frac{W_T - W_p + Q_H}{Q} \quad ...(6.29)$$

6.21 Mixed pressure turbine

In some industries like chemical and material processing, steel, rolling mills, etc. the steam is required at considerably higher pressure and it is also exhausted at a pressure considerably higher than atmosphere. In this case, the utilisation of exhaust steam in a Low Pressure (L.P.) turbine is advantageous. In this arrangement part of High Pressure (H.P.) steam from boiler is supplied to process plant and L.P. steam coming from process plant is supplied to L.P. turbine where it further expands to condenser pressure. The arrangement of such turbine is shown in Fig. 6.59.

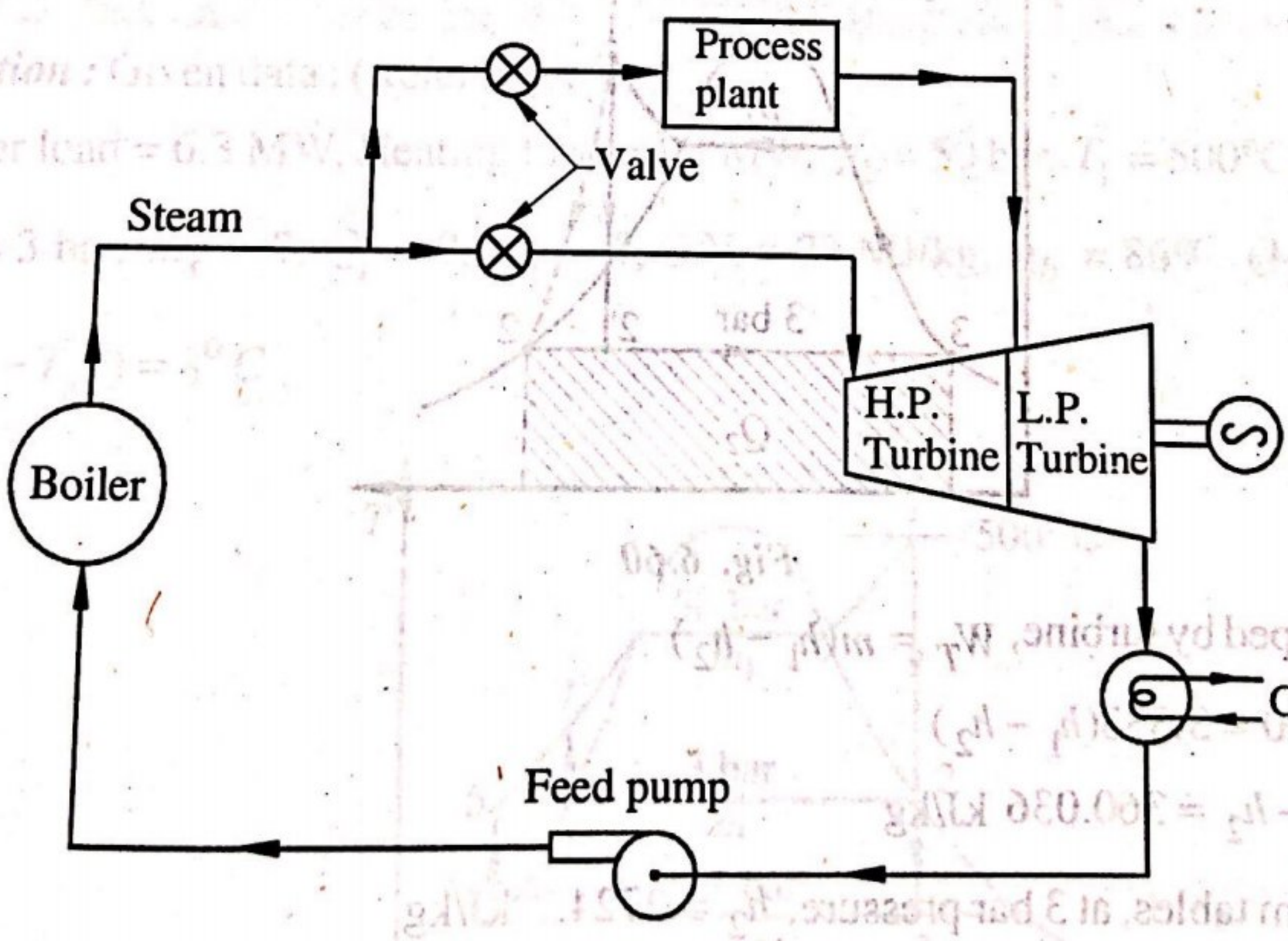


Fig. 6.59 Mixed pressure turbine

The mixed pressure turbine is designed in such a way that live steam (H.P. steam) may be admitted to the turbine at an earlier stage than the first L.P. stage. So we can say that this

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turbine consists of two turbines, one H.P. and the other L.P. turbine but placed generally in the same casing. The steam coming out from last stage H.P. turbine enters the first set of nozzle of the L.P. turbine. If the processing plant is also operating then the L.P. steam is getting mixed with the steam coming out from the last stage of H.P. turbine and after that enters the first set of nozzles of the L.P. turbines, thus the name "*mixed -pressure turbine*". Both H.P. steam and L.P. steam are controlled by centrifugal governor to maintain the speed of the turbine fairly constant irrespective of load. So there are three possible conditions of operation :-

(1) Power developed entirely by L.P. steam,

(2) Power developed entirely by H.P. steam,

(3) Power developed partly by H.P. steam and partly by L.P. steam, thus the name mixed-pressure turbine.

Inplace of process plant, L.P. exhaust steam from reciprocating steam engine may be used in mixed-pressure turbine.

Problem 6.25 *A textile factory requires 12 t/h of steam for process heating at 3 bar saturated and 1200 kW of power, for which a back pressure turbine of 72% internal efficiency is to be used. Determine the steam condition required at inlet of the turbine.*

Solution : Given data (refer Fig. 6.60)

$m = 12\ t/h = (12 \times 10^3) / 3600 = 3.333$ kg/s, $p_2 = 3$ bar, $W_T = 1200$ kW, $\eta = 72\%$,

$p_1 = ?, T_1 = ?$

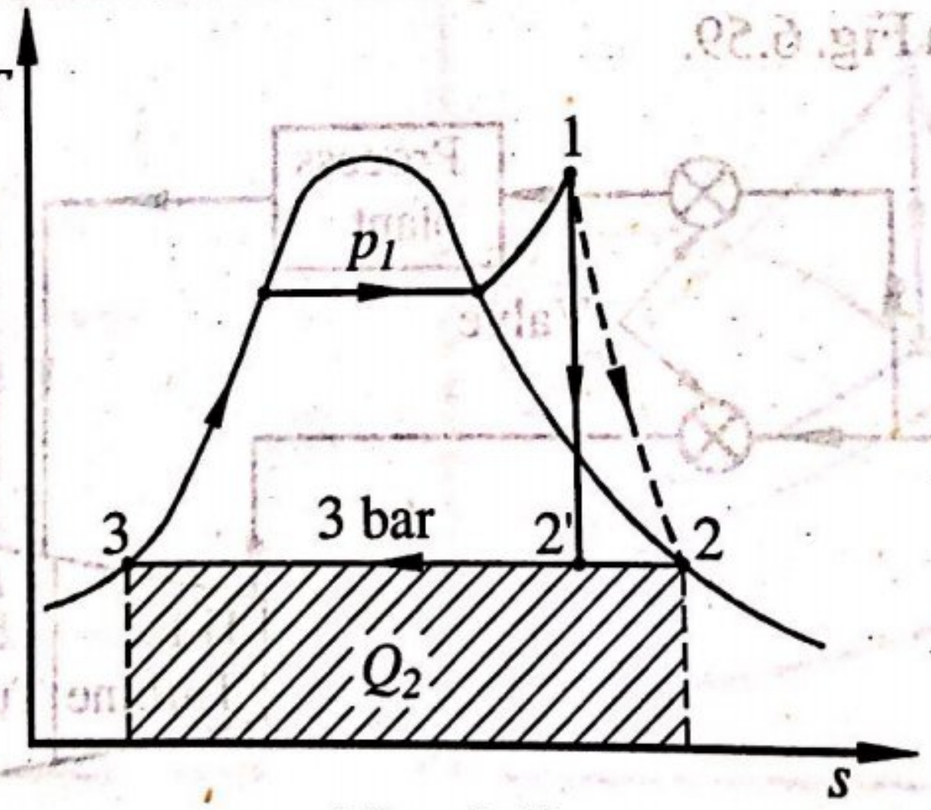


Fig. 6.60

Power developed by turbine, $W_T = m(h_1 - h_2)$

$\therefore 1200 = 3.333(h_1 - h_2)$

$\therefore h_1 - h_2 = 360.036$ kJ/kg

From the steam tables, at 3 bar pressure, $h_2 = 2724.7$ kJ/kg

$\therefore h_1 - 2724.7 = 360.036$

$\therefore h_1 = 3084.736$ kJ/kg

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Now, $h_1 - h_2' = \dfrac{360.036}{0.72} = 500.05\,kJ/kg$

$\therefore h_2' = 3084.736 - 500.05 = 2584.686\,kJ/kg$

$h_2' = h_f + x_2' h_{fg} = 561.5 + x_2' \times 2163.2 = 2584.686$

$\therefore x_2' = 0.9353$

$s_2' = s_f + x_2' s_{fg} = 1.672 + 0.9353 \times 5.319 = 6.647\ kJ/kg\,K$

For isentropic expansion, $s_1 = s_2'$

From the Mollier diagram corresponding to $h_1 = 3084.736$ kJ/kg and $s_1 = 6.647$ kJ/kg K, We get $p_1 = \mathbf{38\ bar}$ and $T_1 = \mathbf{340^oC}$. ***Ans***

Problem 6.26 *In a pass-out turbine (cogeneration plant), the power load is 6.3 MW and the heating load is 1.4 MW. Boiler generates the steam at 50 bar and 500°C and steam is expanded isentropically through a turbine to a condenser at 0.1 bar. The heating load is supplied by extracting steam from the turbine at 3 bar which condensed in the process heater to saturated liquid at 3 bar and then pumped back, to the boiler. Calculate (i) the steam generation capacity of the boiler in t/h, (ii) the heat input to the boiler in kW, (iii) the fuel burning rate of the boiler in t/h if a coal of calorific value 23 MJ/kg is burned, take boiler efficiency as 86%, (iv) the heat rejected to the condenser, (v) the rate of flow of cooling water in the condenser if the temperature rise of water is 8°C. Neglect pump work.*

Solution : Given data : (Refer Fig. 6.61)

Power load = 6.3 MW, Heating load = 1.4 MW, $p_1 = 50$ bar, $T_1 = 500°C$, $p_3 = 0.1$ bar, $p_2 = 3$ bar, $m_s = ?$, $Q_1 = ?$, $m_f = ?$, C.V. = 23 MJ/kg, $\eta_b = 86\%$, $Q_2 = ?$, $m_w = ?$, $(T_{w_2} - T_{w_1}) = 8^o C$.

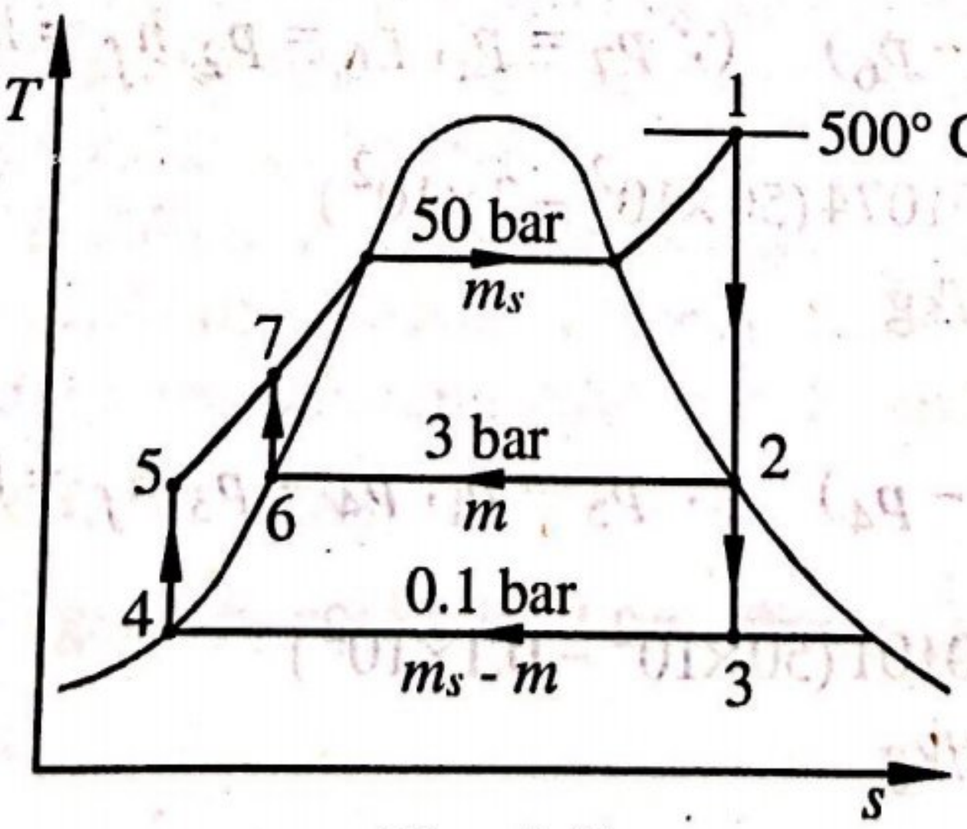


Fig. 6.61

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From superheated steam tables corresponding to $p_1 = 50$ bar and $T_1 = 500°C$,

$$h_1 = 3433.7 \text{ kJ/kg},\ s_1 = 6.977 \text{ kJ/kg K} = s_2 = s_3$$

$$s_2 = s_f + x_2 s_{fg} = 1.672 + x_2 \times 5.319 = 6.977$$

$$\therefore x_2 = 0.997 \cong 1$$

From steam tables corresponding to $p_2 = 3$ bar

$$h_2 = h_g = 2724.7 \text{ kJ/kg},\ h_6 = h_f = 561.5 \text{ kJ/kg}$$

If m is the rate of steam extraction for process heating.

$$m(h_2 - h_6) = 1.4 \times 10^3$$

$$\therefore m(2724.7 - 561.5) = 1.4 \times 10^3$$

$$\therefore m = 0.6472 \text{ kg/s} = 2329.92 \text{ kg/h}$$

$$s_1 = s_3 = s_f + x_3 s_{fg} = 0.649 + x_3 \times 7.502 = 6.977$$

$$\therefore x_3 = 0.8435$$

We have $h_3 = h_f + x_3 h_{fg} = 191.8 + 0.8435 \times 2392.9$

$$= 2210.211 \text{ kJ/kg}$$

Total turbine work, $W_T = m_s(h_1 - h_2) + (m_s - m)(h_2 - h_3)$

$$6.3 \times 10^3 = m_s(3433.7 - 2724.7) + (m_s - 0.6472)(2724.7 - 2210.211)$$

$$= m_s \times 709 + m_s \times 514.489 - 332.977$$

$$\therefore m_s = 5.42 \text{ kg/s} = 19516.9 \text{ kg/h} = \mathbf{19.5169 \text{ t/h}} \quad \textit{Ans}$$

Enthalpy at state point 7

$$h_7 = h_f + v_f(p_7 - p_6) \quad (\because p_7 = p_1,\ p_6 = p_2, h_f = h_6)$$

$$= 561.5 + 0.001074(50 \times 10^2 - 3 \times 10^2)$$

$$= 566.5478 \text{ kJ/kg}$$

Enthalpy at state point 5

$$h_5 = h_f + v_f(p_5 - p_4) \quad (\because p_5 = p_1,\ p_4 = p_3, h_f = h_{f_4})$$

$$= 191.8 + 0.00101(50 \times 10^2 - 0.1 \times 10^2)$$

$$= 196.8399 \text{ kJ/kg}$$

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Heat supplied to the boiler,

$$Q_1 = (m_s - m)(h_1 - h_5) + m(h_1 - h_7)$$
$$= (5.42 - 0.6472)(3433.7 - 196.8399) + 0.6472(3433.7 - 566.5478)$$
$$= 15448.886 + 1855.621 = 17304.507 \text{ kJ/s}$$
$$= \mathbf{17.3045\ MW}$$ ***Ans***

Boiler efficiency is given by

$$\eta_b = \frac{Q_1}{m_f \times C.V.}$$

$$\therefore 0.86 = \frac{17.3045}{m_f \times 23}$$

$$\therefore m_f = 0.8748 \text{ kg/s} = 3149.28 \text{ kg/h} = \mathbf{3.149\ t/h}$$ ***Ans***

Heat rejected to condenser,

$$Q_2 = (m_s - m)(h_3 - h_4) = (5.42 - 0.6472)(2210.211 - 191.8)$$
$$= 9633.472 \text{ kJ/s or kW}$$
$$= \mathbf{9.63347\ MW}$$ ***Ans***

Let m_w = Cooling water flow rate in the condenser.

$$Q_2 = m_w C_p (T_{w_2} - T_{w_1})$$

$$\therefore 9633.472 = m_w \times 4.187 \times 8$$

$$\therefore m_w = 287.6 \text{ kg/s} = \mathbf{0.2876\ m^3/s}$$ ***Ans***

Problem 6.27 *In a combined power and process plant the boiler generates 12600 kg/h of steam at a pressure of 15 bar and temperature 250°C. A part of the steam goes to a process heater which consumes 125 kW, the steam leaving the process heater 0.95 dry at 15 bar being throttled to 3 bar. The remaining steam flows through an H.P. turbine which exhausts at a pressure of 3 bar. The exhaust steam mixes with the process steam before entering the L.P. turbine which develops 1250 kW. At the exhaust, the pressure is 0.1 bar and the steam is 0.88 dry. Draw the line and T-s diagrams of the plant and determine (i) the steam quality at the exhaust of the H.P. turbine, (ii) the power developed by the H.P. turbine, and (iii) the isentropic efficiency of the H.P. turbine.*

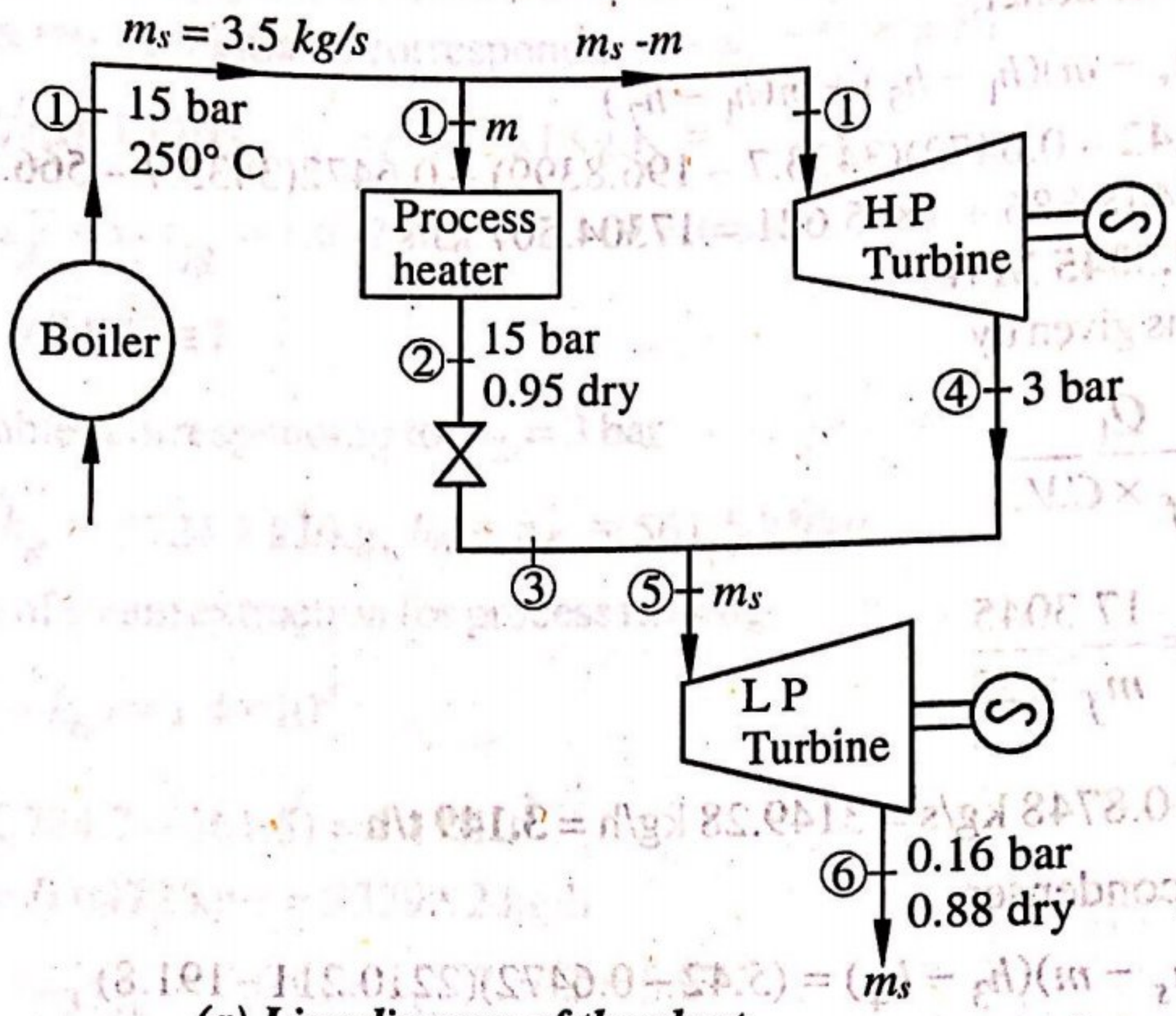
Solution : Given data (Refer Fig. 6.62)

$m_s = 12600 \text{ kg/h} = 3.5 \text{ kg/s}$, $p_1 = 15 \text{ bar}$, $T_1 = 250°C$, $Q_H = 125 \text{ kW}$, $p_4 = 3 \text{ bar}$,

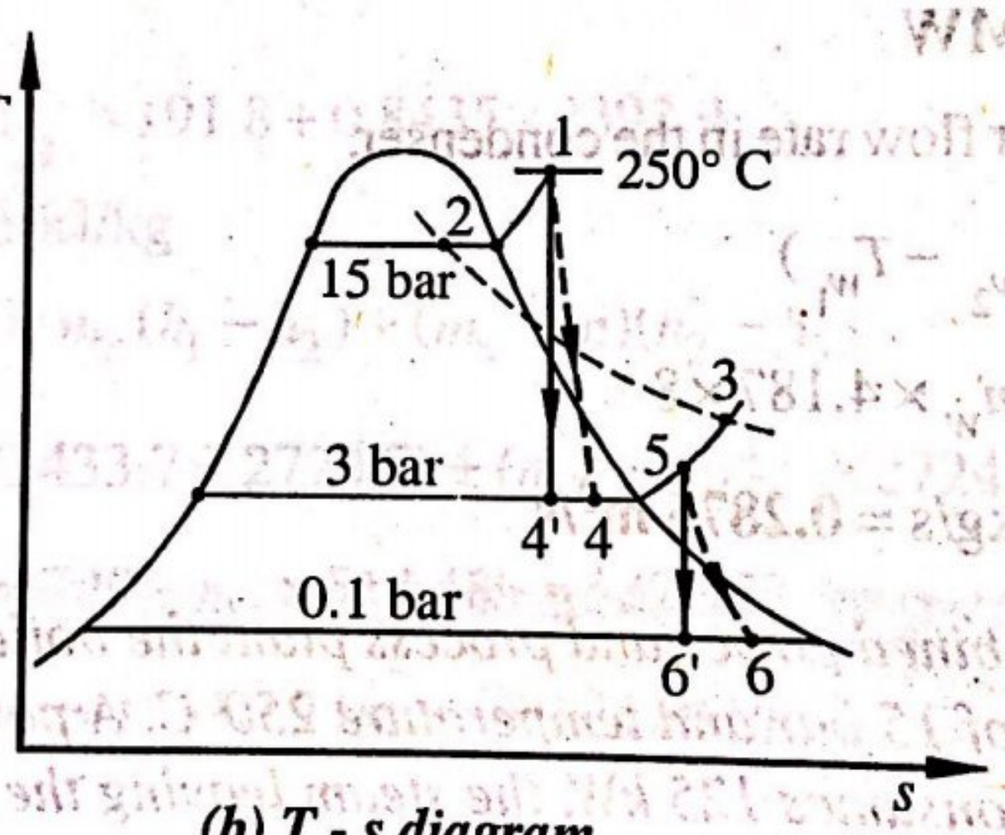
$p_6 = 0.1 \text{ bar}$, $x_6 = 0.88$

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(a) Line diagram of the plant



(b) T - s diagram

Fig. 6.62

(i) The steam quality at the exhaust of the H.P. turbine :

From the superheated steam tables corresponding to $p_1 = 15$ bar and $T_1 = 250°C$

$h_1 = 2923.5$ kJ/kg, $s_1 = 6.710$ kJ/kg K

From the saturated steam tables corresponding to $p_1 = p_2 = 15$ bar

$h_f = 844.6$ kJ/kg, $h_{fg} = 1945.3$ kJ/kg

$\therefore h_2 = h_f + x_2 h_{fg} = 844.6 + 0.95 \times 1945.3 = 2692.635$ kJ/kg

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Throttling is constant enthalpy process,

$$\therefore h_2 = h_3$$

Work done by L.P. turbine = 1250 = $m_s (h_5 - h_6)$

$$\therefore h_5 - h_6 = \frac{1250}{3.5} = 357.143 \text{ kJ/kg}$$

From steam tables corresponding to $p_6 = 0.1$ bar

$$h_f = 191.8 \text{ kJ/kg}, \; h_{fg} = 2392.9 \text{ kJ/kg}$$

$$\therefore h_6 = h_f + x_6 h_{fg} = 191.8 + 0.88 \times 2392.9 = 2297.552 \text{ kJ/kg}$$

$$\therefore h_5 - 2297.552 = 357.143$$

$$\therefore h_5 = 2654.695 \text{ kJ/kg}$$

From steam tables corresponding to $p_4 = 3$ bar

$$h_f = 561.5 \text{ kJ/kg}, \; h_{fg} = 2163.2 \text{ kJ/kg},$$

$$s_f = 1.672 \text{ kJ/kg K}, \; s_{fg} = 5.319 \text{ kJ/kg}$$

We have, $s_1 = s_4' = s_f + x_4' s_{fg}$

$$\therefore 6.710 = 1.672 + x_4' \times 5.319$$

$$\therefore x_4' = 0.947$$

Heat required for process heating is given by

$$Q_H = m(h_1 - h_2); \; 125 = m(2923.5 - 2692.635)$$

$$\therefore m = 0.5414 \text{ kg/s}$$

$$\therefore (m_s - m) = 3.5 - 0.5415 = 2.9586 \text{ kg/s}$$

From the line diagram,

$$mh_3 + (m_s - m)h_4 = m_s h_5$$

$$\therefore 0.5414 \times 2692.635 + 2.9586 \times h_4 = 3.5 \times 2654.695$$

$$\therefore h_4 = 2647.752 \text{ kJ/kg}$$

Also $h_4 = h_f + x_4 h_{fg}$

$$2647.752 = 561.5 + x_4 \times 2163.2$$

$$\therefore x_4 = \mathbf{0.9644}$$ **Ans**

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(ii) The power developed by H.P. Turbine :

$$(W_T)_{HP} = (m_s - m)(h_1 - h_4) = 2.9586(2923.5 - 2647.752)$$

$$= \mathbf{815.828\ kW} \quad \textbf{Ans}$$

(iii) The isentropic efficiency of the H.P. Turbine :

$$(\eta_{isen})_{HP} = \frac{\text{Actual heat drop}}{\text{Isentropic heat drop}} = \frac{h_1 - h_4}{h_1 - h_4'}$$

$$= \frac{2923.5 - 2647.752}{2923.5 - 2610.05} = 0.87972 = \mathbf{87.972\%} \quad \textbf{Ans}$$

Exercise

6.1 Define steam turbine. How does energy conversion take place in steam turbine?
6.2 Explain in brief advantages and disadvantages of steam turbine.
6.3 Explain in brief principle and operation of steam turbine.
6.4 Give detailed classification of steam turbines.
6.5 What do you understand by impulse and reaction turbines?
6.6 How is turbines classified according to direction of steam flow?
6.7 What is the difference between condensing and non-condensing steam turbines?
6.8 What do you understand by regenerative steam turbine?
6.9 What do you understand by compounding of steam turbine? State the different methods of compounding of steam turbine and explain with neat sketch velocity compounded impulse steam turbine.
6.10 Explain pressure compounded impulse steam turbine with net sketch and give advantages and disadvantages of the same.
6.11 Explain pressure-velocity compounded impulse turbine with neat sketch.
6.12 Define : (i) Blade efficiency, (ii) Stage efficiency, (iii) Blade speed ratio.
6.13 Explain the effect of blade friction on velocity diagram of simple impulse steam turbine.
6.14 Define blade efficiency and hence derive an expression for maximum blade efficiency for a single stage impulse steam turbine.
6.15 Explain in brief condition for maximum efficiency of impulse turbine.
6.16 Explain reaction steam turbine showing variation of pressure and velocity along the axis of turbine.
6.17 Define degree of reaction and show that degree of reaction $R = \frac{V_f}{2V_b}(\cot\phi - \cot\theta)$.
6.18 Explain Parson's reaction turbine. Why is it called 50% reaction turbine?
6.19 Derive an expression for maximum efficiency of Parson's reaction turbine.
6.20 Give detailed comparison of impulse and reaction steam turbines.

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6.21 Explain "Reheat factor" and "Internal efficiency". Derive the relation between stage efficiency, internal efficiency and reheat factor.

6.22 Explain impulse blade sections with neat sketch. Also state limitations of plate blade.

6.23 Explain impulse-reaction turbine blade section with neat sketch.

6.24 List the different losses occurring in steam turbines. Explain the concept of any one with neat sketch.

6.25 What is compounding of impulse turbine ? List main types of compounding. Explain working principle of pressure compounded impulse turbine.

6.26 For an impulse turbine, explain the following terms and also obtain expressions for them:-
(i) Power (ii) Axial thrust (iii) Blade efficiency

6.27 Why is governing of steam turbine required?

6.28 State the various methods of governing of a steam turbine and explain any one of them in detail.

6.29 Explain throttle governing with neat sketch.

6.30 Explain with neat sketch nozzle control governing of steam turbine.

6.31 Explain with neat sketch by-pass method of governing of steam turbine.

6.32 Compare throttle control governing with nozzle control governing of steam turbine.

6.33 Explain throttle control governing of steam turbine. Compare it with nozzle control governing.

6.34 When bypass governing is required in steam power plant ? Explain the working of bypass governing.

6.35 Why nozzle control governing is not used in reaction turbine ?

6.36 What is Willans line ?

6.37 Justify the statement "Back pressure and pass-out turbines are advantageous for power and process work than generating steam for both work separately".

6.38 Why the combined power and process heating plant is more efficient ?

6.39 Explain with schematic diagram, the working, utility and importance of back pressure turbine.

6.40 Explain with schematic diagram the importance, working and application of back pressure turbine. When and why the steam is de-superheated ?

6.41 Write short notes on working principle and application of "pass-out" turbine.

6.42 Explain pass-out turbine with T-s and h-s diagrams.

6.43 Explain pass-out turbine with neat sketch.

6.44 In what conditions, mixed pressure turbine is needed ? Explain with diagram the arrangement and working of mixed pressure turbine.

6.45 What is the function of steam accumulator ? Also explain the operation of the steam accumulator.

6.46 Explain with a neat sketch a steam accumulator.

6.47 Write short notes on the Labyrinth glands and packing used in steam turbine.

6.48 Explain with neat sketches, any two methods of blade fixation on the turbine rotor.

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6.49 Discuss briefly the various losses occurring in steam turbines.

6.50 Explain in brief (i) de Laval blade attachment, (ii) Serrated blade root attachment.

6.51 Explain in brief : Disc friction and blade windage losses in steam turbine.

Unsolved Numerical Problems

6.1 Steam issues from nozzles, inclined at an angle 24° to the direction of motion, with a velocity 1000 m/s on to the blades of a single stage impulse turbine. The velocity of the blades is 400 m/s and the blades are equiangular. The steam flow rate is 4000 kg/h. Determine (i) inlet angle of the blades, (ii) force exerted on the blades in the direction of their motion (iii) power developed by the rotor. Assume frictionless flow in blades and steam enters the blade without shock.

[**Ans.** (i) 39°, (ii) 1.155 kN, (iii) 462 kW]

6.2 Steam issues from the nozzles of a De Laval turbine with a velocity of 720 m/s. The nozzle angle is 16°, the mean blade velocity is 180 m/s and the blade angle at outlet is 25°. The mass of steam flowing through turbine per second is 1 kg. The blade velocity co-efficient is 0.72. Determine : (i) power of the wheel, (ii) blade efficiency (iii) the energy lost in the blades. [**Ans.** (i) 157.5 kW (ii) 60.75%, (iii) 72.84 kJ/kg]

6.3 In a stage of an impulse turbine provided with a single row wheel, the mean diameter of the blade ring is 100 cm and the speed of rotation is 3000 rpm. The steam issues from the nozzles with a velocity of 300 m/s and the nozzle angle is 20°. The rotor blades are equiangular. If the friction loss is 35% of kinetic energy corresponding to relative velocity at inlet, find the power developed by the turbine. The axial thrust is 150 N.

[**Ans.** 262 kW]

6.4 In a single stage impulse turbine, the mean diameter of the blade ring is 105 cm and the speed of rotation is 3000 rpm. The steam issues from the nozzles with an angle of 18°. The ratio of the blade speed to steam speed at entry to blade is 0.42. The blade velocity co-efficient is 0.84. The outlet angle of the moving blade is 3° less than the inlet angle. Determine : (i) power developed, (ii) blade efficiency, (iii) axial thrust.

[**Ans.** (i) 64.16 kW/kg of steam, (ii) 83.23%, (iii) 59.10 N/kg of steam]

6.5 In simple impulse turbine, the velocity of steam leaving the nozzles is 1200 m/s and the nozzle angle 20°. The blade velocity is 375 m/s and the blade velocity co-efficient is 0.75. Mass flow of steam is 0.5 kg/s and blades are symmetrical. Determine : (i) Blade inlet angle, (ii) driving force on the wheel, (iii) axial thrust on the wheel, and (iv) power developed by the turbine. [**Ans.** (i) 29°, (ii) 660 N, (iii) 50 N, (iv) 247.5 kW]

6.6 A simple impulse turbine has a mean blade ring diameter of 2.5 m and runs at 3000 rpm. The nozzle angle is 20° and ratio of blade velocity to steam velocity is 0.4. The blade friction factor is 0.8 and blade exit angle is 3° less than that at inlet. Steam flow rate is 36000 kg/h. Draw velocity diagram for moving blade and determine : (i) power developed. (ii) blade efficiency, and (iii) steam consumption in kg/kWh.

[**Ans.** (i) 3800 kW, (ii) 78.7%, (iii) 9.47 kg/kWh]

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6.7 In stage of 50% reaction turbine, the steam consumption is 15400 kg/h and it runs at 300 rpm. The discharge blade tip angles are 20° both for moving and fixed blades. The axial velocity of flow is 0.72 times the blade velocity. The pressure of steam at a certain pair is 1.9 bar, its dryness 0.93 and power developed by the pair is 3.5 kW. Take the tip leakage steam as 8% but neglect blade thickness. Find the drum diameter and blade height. [**Ans.** D=1.032 m, h= 78 mm]

6.8 A reaction turbine with a mean blade diameter of 100 cm runs at a speed of 3000 rpm. The blades are designed with exit angles of 50° and inlet angles of 30°. If the turbine is supplied with steam at the rate of 20 kg/s and gross efficiency is 85%, determine : (i) power output of the stage, (ii) specific enthalpy drop in the stage, and (iii) percentage increase in relative velocity in the moving blades due to steam expansion.

[**Ans.** (i) 1413 kW, (ii) 83.1kJ/kg of steam, (iii) 53.2%]

6.9 A reaction steam turbine runs at 300 rpm and its steam consumption is 4.58 kg/s. The pressure of steam at a certain pair is 1.76 bar (abs.) and its dryness fraction is 0.9 and the power developed by the pair is 3.31 kW. The discharge blade tip angle both for fixed and moving blade is 20° and the axial velocity of flow is 0.72 of the mean moving blade velocity. Determine the drum diameter and blade height. Take the tip leakage as 8%, but neglect area blocked by blade thickness. [**Ans.** (i) D=1.03 m, (ii) h=0.1 m]

6.10 The following particulars refer to a stage of an impulse reaction turbine : Mean diameter = 680 mm, turbine speed = 3000 rpm, mass flow rate of steam = 13.5 kg/s. steam velocity at exit from fixed blades = 143.7 m/s, blade outlet angle = 20°, carryover co-efficient = 0.74, expansion efficiency in the blade channels = 0.92. Determine : (i) the power developed in the stage, (ii) the gross efficiency. [**Ans.** (i) 236.46 kW, (ii) 89%]

6.11 At a stage in a reaction turbine, the mean drum diameter is 1.5 m and the mean blade height is 150 mm. The blade tip angles of fixed and moving blades at inlet and exit are 30° and 20° respectively. The turbine is supplied with dry saturated steam at the rate 200 kg/s and pressure of 15 bar. The stage efficiency is 85%. Determine : (i) the axial velocity of flow, (ii) the blade speed, (iii) the power output of the stage, (iv) the specific enthalpy drop in this stage, (v) the percentage of increase in relative velocity in the moving blades due to expansion in these blades.

[**Ans.** (i) 34.47 m/s, (ii) 34.99 m/s, (iii) 1080.42 kW, (iv) 6355 kJ/kg, (v) 46.18%]

6.12 A certain chemical plant requires 8 t/h of steam for process heating at 2 bar dry saturated and 800 kW of power, for which a back pressure turbine of 70% internal efficiency is to be used. Determine the steam condition required at inlet of the turbine.

[**Ans :** 27 bar, 330°C]

6.13 A 12 MW steam turbine operates with steam at 40 bar and 400°C at the inlet and exhaust at 0.06 bar. 12,000 kg/h of steam at 2 bar are to be extracted for process work. The turbine has 76% isentropic efficiency throughout. Find the boiler capacity required.

[**Ans :** 13.9206 kg/s]

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Objective Type Questions

1. In case of impulse steam turbine
   (a) there is pressure drop in nozzles and moving blades
   (b) there is pressure drop in nozzles only
   (c) there is pressure drop in moving blades only
   (d) all of the above
2. De-Laval turbine is an example of
   (a) simple impulse turbine
   (b) pressure compounded impulse turbine
   (c) velocity compounded impulse turbine
   (d) reaction turbine
3. In case of reaction steam turbine
   (a) there is pressure drop in fixed blades only
   (b) there is pressure drop in moving blades only
   (c) there is pressure drop in fixed and moving blades
   (d) all of the above
4. The following statement is not true for reaction steam turbines
   (a) steam expands in fixed as well as in moving blades.
   (b) It has higher diagram efficiency than impulse turbine
   (c) It occupies relatively more space
   (d) It's blade channels area is constant
5. In De-Laval steam turbine
   (a) the pressure in the turbine rotor is higher than the condenser pressure.
   (b) the pressure in the turbine rotor is less than the condenser pressure.
   (c) the pressure in the turbine rotor gradually decreases from inlet to exit to condenser pressure.
   (d) the pressure in the turbine rotor is approximately equal to condenser pressure.
6. In regenerative steam turbine
   (a) a bled steam is used for process heating
   (b) bled steam is used for feed water heating
   (c) exhaust steam from turbine is used for process heating
   (d) all of the above
7. In impulse turbines, when friction is neglected, the relative velocity of steam at outlet tip of the blade is _____ the relative velocity of steam at inlet tip of the blade.
   (a) greater than (b) less than
   (c) equal to (d) none of the above
8. Compounding of steam turbine is the method of
   (a) reducing the rotational speed of the turbine
   (b) increasing efficiency of the turbine

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(c) increasing the rotational speed of the turbine
(d) reducing steam consumption of the turbine

9. Rateau steam turbine is
(a) pressure - velocity compounded impulse steam turbine
(b) velocity compounded impulse steam turbine
(c) pressure compounded impulse steam turbine
(d) reaction steam turbine

10. Curtis steam turbine is
(a) pressure-velocity compounded impulse steam turbine
(b) velocity compounded impulse steam turbine
(c) pressure compounded impulse steam turbine
(d) reaction steam turbine

11. Blade or diagram efficiency is given by

(a) $\frac{2V_w \cdot V_b}{V_1}$ (b) $\frac{V_w V_b}{V_1^2}$ (c) $\frac{2V_w^2}{V_1^2}$ (d) $\frac{2V_w V_b}{V_1^2}$

12. Axial thrust on rotor of steam turbine is given by

(a) $m(V_{f1} - V_{f2})$ (b) $m(V_1 - V_2)$ (c) $m(V_{f1} + V_{f2})$ (d) $m(V_1 + V_2)$

13. Condition for maximum diagram efficiency of single stage impulse turbine is

(a) $\rho = \frac{\cos^2 \alpha}{2}$ (b) $\rho = \cos\alpha$ (c) $\rho = \frac{\cos\alpha}{2}$ (d) $\rho = \cos^2 \alpha$

14. Maximum blade efficiency of single stage impulse turbine is given by

(a) $(1 - KB)\frac{\cos^2 \alpha}{2}$ (b) $(1 - KB)\frac{\cos\alpha}{2}$ (c) $(1 + KB)\frac{\cos\alpha}{2}$ (d) $(1 + KB)\frac{\cos^2 \alpha}{2}$

15. Degree of reaction as referred to steam turbine is given by

(a) $\frac{\Delta h_f}{\Delta h_f + \Delta h_m}$ (b) $\frac{\Delta h_m}{\Delta h_m + \Delta h_f}$ (c) $\frac{\Delta h_m}{\Delta h_f}$ (d) $\frac{\Delta h_f}{\Delta h_m}$

16. Following is a particular case of reaction turbine in which the degree of reaction is 0.5.
(a) Curtis turbine (b) De-Lavel turbine
(c) Rateau turbine (d) Parson's turbine

17. Following is the condition for maximum blade efficiency of Parson's reaction turbine.

(a) $\rho = \cos\alpha$ (b) $\rho = \cos^2 \alpha$ (c) $\rho = \frac{\cos\alpha}{2}$ (d) $\rho = \frac{\cos^2 \alpha}{2}$

18. The maximum efficiency for Parson's reaction turbine is given by

(a) $\frac{\cos^2 \alpha}{1 + \cos^2 \alpha}$ (b) $\frac{2\cos^2 \alpha}{1 - \cos^2 \alpha}$ (c) $\frac{2\cos^2 \alpha}{1 + \cos^2 \alpha}$ (d) $\frac{\cos^2 \alpha}{1 - \cos^2 \alpha}$

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19. Steam turbines are governed by which of the following methods ?
(a) Throttle governing (b) Nozzle governing
(c) By-pass governing (d) All of the above

20. _____ method of steam turbine governing is generally used in small turbines.
(a) Throttle (b) Nozzle (c) By-pass (d) all of the above

21. In which method of governing, partial admission losses are more
(a) Throttle (b) Nozzle (c) By pass (d) all of the above

22. _____ method of steam turbine governing is generally not suitable for reaction turbine.
(a) Throttle (b) Nozzle (c) By pass (d) all of the above

23. Following is the most efficient method of steam turbine governing
(a) Throttle (b) Nozzle (c) By pass (d) all of the above

24. In which of the following turbine, condenser is not used ?
(a) pass-out turbine (b) back pressure turbine
(c) mixed pressure turbine (d) none of the above

25. In _____ turbine, de-superheater is used.
(a) pass-out turbine (b) back pressure turbine
(c) mixed pressure turbine (d) none of the above

26. Why is the cogeneration of power and process heat advantageous ?
(a) heat rejected to condenser is less (b) plant efficiency is high
(c) two separate steam generators are not needed (d) all of the above

27. The following turbine is also called extraction turbine
(a) pass-out turbine (b) back pressure turbine
(c) mixed-pressure turbine (d) all of the above

28. In pass-out turbine
(a) steam is supplied with different pressures to the turbine
(b) steam supplied to the turbine is condensed in condenser.
(c) a certain quantity of steam is continuously extracted from the turbine at an intermediate stage for process heating.
(d) none of the above

29. In mixed-pressure turbine
(a) steam is supplied with different pressures to the different stages of turbine.
(b) steam supplied to the turbine is not condensed in condenser.
(c) steam is extracted from turbine for process heating
(d) all of the above

Answers

1. (b) **2.** (a) **3.** (c) **4.** (d) **5.** (d) **6.** (b) **7.** (c) **8.** (a) **9.** (c) **10.** (b)
11. (d) **12.** (a) **13.** (c) **14.** (d) **15.** (b) **16.** (d) **17.** (a) **18.** (c) **19.** (d) **20.** (a)
21. (b) **22.** (b) **23.** (c) **24.** (b) **25.** (b) **26.** (d) **27.** (a) **28.** (c) **29.** (a)

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GTU Study Materials ☑

Chapter 7
Condensers and Cooling Towers

7.1 Introduction
7.2 Types of condensers
7.3 Jet condensers
7.3.1 Low level-parallel flow jet condenser
7.3.2 Low level-counter flow jet condenser
7.3.3 High level jet condenser
7.3.4 Ejector condenser
7.3.5 Advantages and disadvantages of jet condensers
7.4 Surface condensers
7.4.1 Double pass surface condenser
7.4.2 Down flow surface condenser
7.4.3 Central flow surface condenser
7.4.4 Evaporative condenser
7.4.5 Advantages and disadvantages of surface condensers
7.4.6 Comparison between jet and surface condensers
7.4.7 Requirements of a good surface condenser
7.5 Sources of air in condenser
7.6 Effects of air leakage
7.7 Methods of obtaining maximum vacuum in condenser
7.8 Edward air pump
7.9 Dalton's law of partial pressure
7.10 Vacuum and condenser efficiency
7.11 Mass of cooling water required for condensation of steam
7.12 Necessity of cooling ponds and cooling towers
7.13 Condenser water cooling systems
7.14 Types of cooling towers
7.14.1 Natural draught cooling towers
7.14.2 Forced draught cooling tower
7.14.3 Induced draught counter flow tower
7.14.4 Induced draught cross-flow Tower
7.14.5 Comparison between natural and mechanical draught cooling tower
7.15 Cooling ponds

7.1 Introduction

Condenser is a heat exchanger wherein steam is condensed either in direct contact with cooling water or indirect contact with cooling water through a heat transfer medium

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separating them. A steam condenser has the following two objects :

(1) The main object is to creates a low back pressure (vacuum) for the turbine exhaust so as to obtain the maximum possible energy from steam and thus to secure a high efficiency.

(2) The secondary object is to condense the exhaust steam from the turbine and thus recover the high-quality feedwater for reuse in the cycle.

Fig. 7.1 shows p-v diagram of steam turbine with and without condenser. The shaded area in Fig. 7.1 shows the increase in work obtained by fitting a condenser to a non-condensing steam turbine. The thermal efficiency of a condensing unit therefore is higher than that of non-condensing unit for the same available steam.

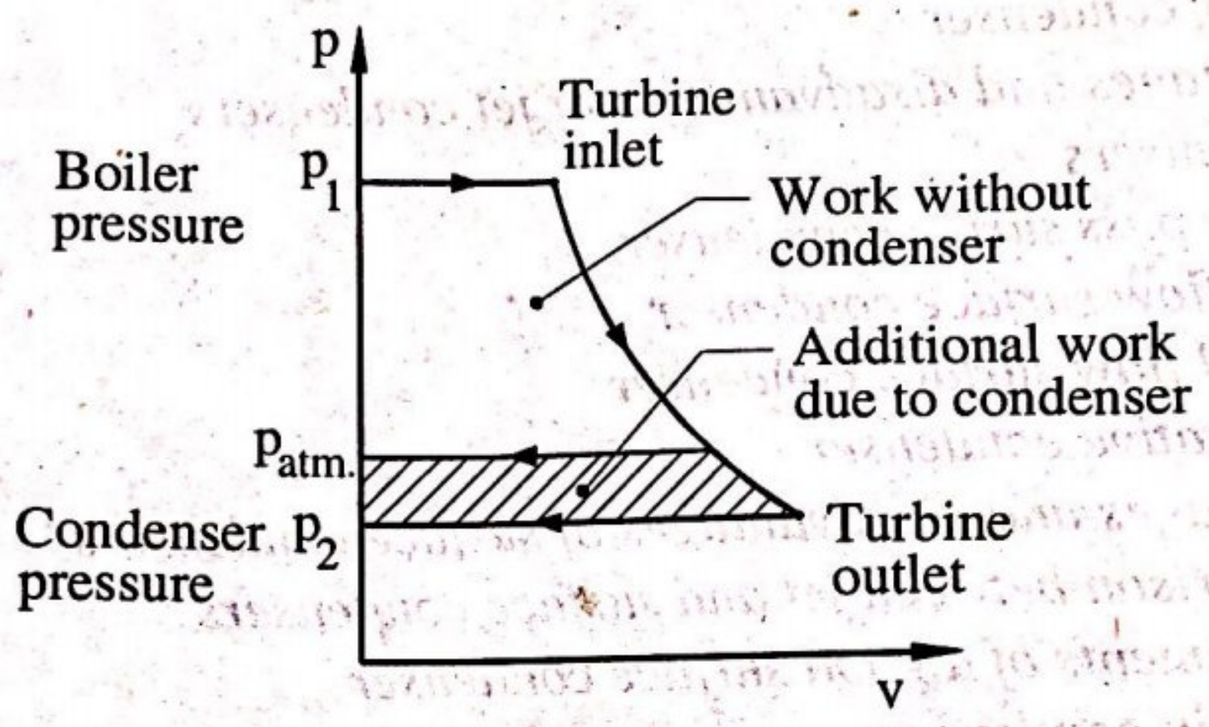


Fig. 7.1 Use of condenser

- **Advantages of condenser :**

(1) It increases work done per kg of steam due to increasing expansion ratio of steam.

(2) It increases thermal efficiency of the plant, hence the power plant size reduces for given output.

(3) The condensed steam provides a source of good pure feed-water to the boiler and reduces the water softening plant capacity to a considerable extent.

(4) The temperature of condensate is higher than that of fresh water. Therefore the amount of heat supplied per *kg* of steam is reduced.

(5) It reduces corrosion of boiler tubes.

- **Elements of steam condensing plant :**

The arrangement of steam condensing plant is shown in Fig. 7.2. The different components of steam condensing plant are given below :

(1) Condenser,
(2) Supply of cooling water,
(3) Condenser cooling water pump,
(4) Condensate extraction pump,
(5) Hot-well,
(6) Boiler feed pump,
(7) Air extraction pump,
(8) Cooling tower,
(9) Make up water pump.

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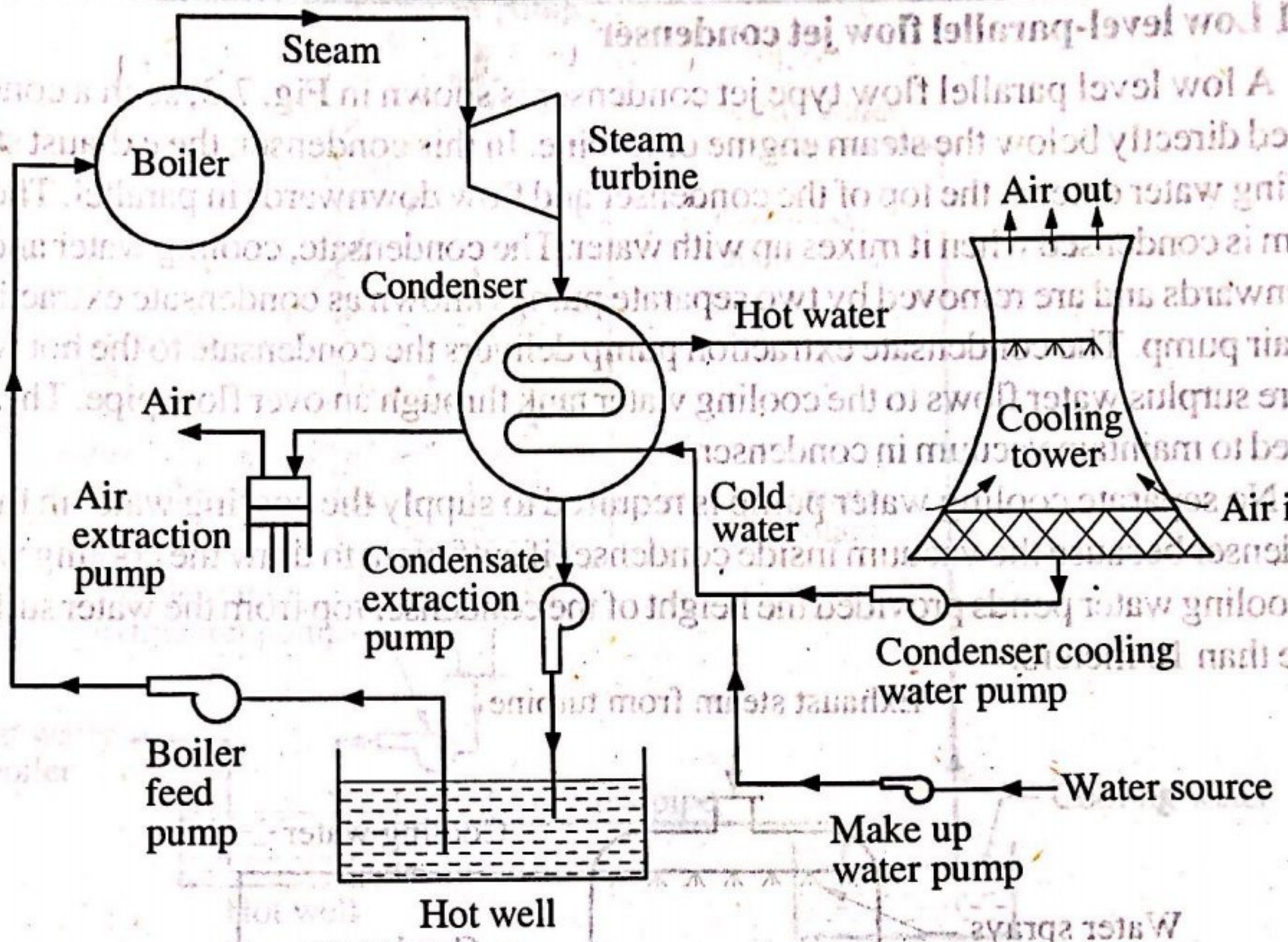


Fig. 7.2 Steam condensing plant

7.2 Types of condensers [Dec. '13]

The two main types of condensers are

(1) Jet condensers

(2) Surface condensers.

In ***jet condensers***, the exhaust steam and cooling water are mixed with each other and the heat transfer from steam to water is by direct contact.

In ***surface condensers***, the exhaust steam and cooling water do not mix with each other, the water being circulated through a nest of tubes and the exhaust steam flows across the tubes, the heat transfer is through tubes. A surface condenser is most commonly used because the condensate obtained is not thrown as a waste but returned to the boiler.

7.3 Jet condensers

The jet condensers may be further classified on the basis of the direction of flow of the condensate and the arrangement of the tubing system, into the following types :

(1) Low level jet condensers

(a) Parallel flow type, (b) Counter flow type

(2) High level jet condensers

(3) Ejector jet condensers.

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7.3.1 Low level-parallel flow jet condenser

A low level parallel flow type jet condenser is shown in Fig. 7.3, such a condenser is placed directly below the steam engine or turbine. In this condenser, the exhaust steam and cooling water enter at the top of the condenser and flow downwards in parallel. The exhaust steam is condensed when it mixes up with water. The condensate, cooling water and air flow downwards and are removed by two separate pumps known as condensate extraction pump and air pump. The condensate extraction pump delivers the condensate to the hot well, from where surplus water flows to the cooling water tank through an over flow pipe. The air pump is used to maintain vacuum in condenser.

No separate cooling water pump is required to supply the cooling water in this type of condenser because the vacuum inside condenser if sufficient to draw the cooling water from the cooling water ponds provided the height of the condenser top from the water surface is not more than 10 meters.

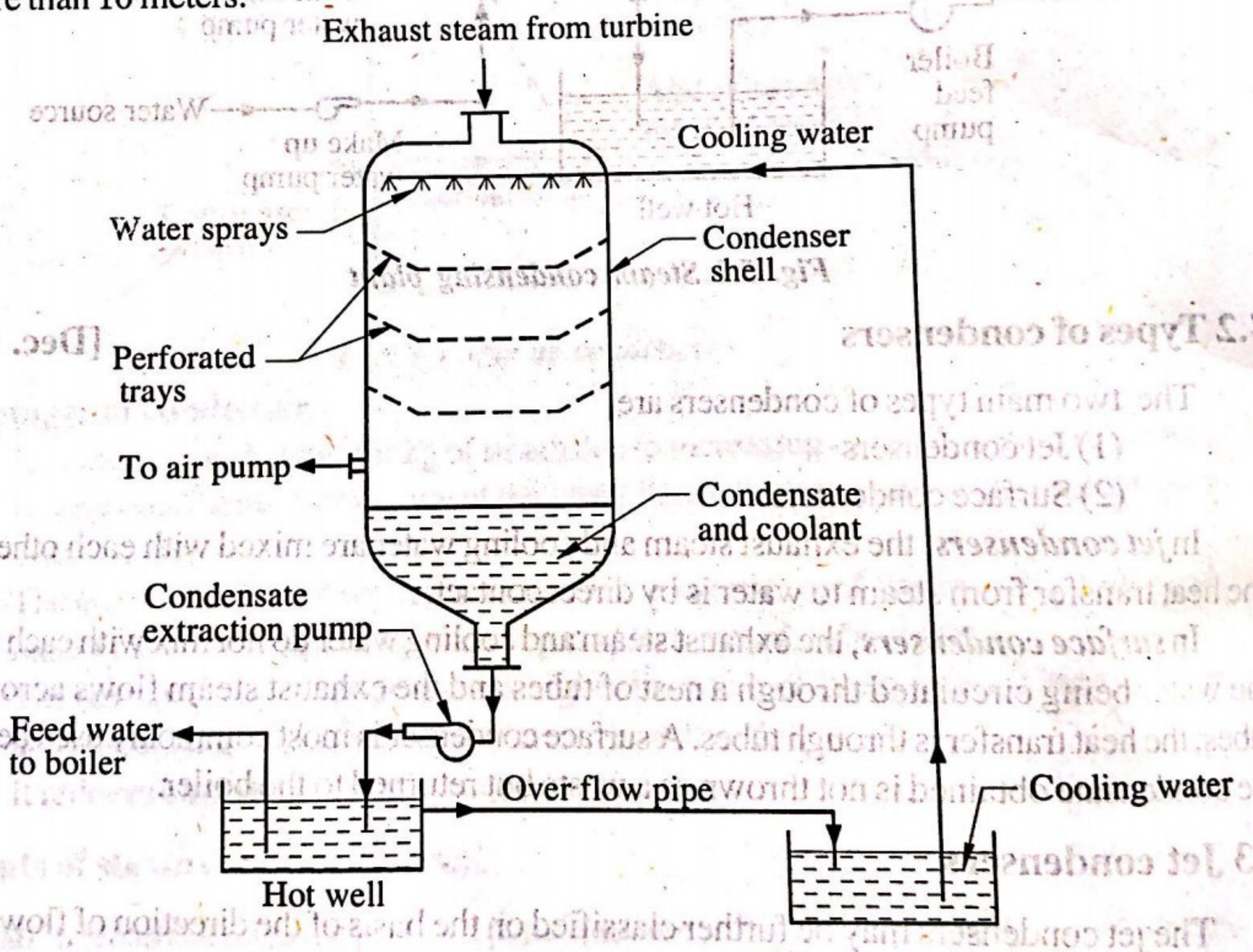


Fig. 7.3. Low level parallel flow jet condenser

7.3.2 Low level-counter flow jet condenser

The arrangement of low level counter flow jet condenser is shown in Fig. 7.4. In this type of condenser, the exhaust steam enters at the bottom, flows upwards and meets the downcoming cooling water. The air pump is placed at the top of the condenser shell. The exhaust steam is condensed when it mixes up with water. Other arrangements are similar to parallel flow jet condenser.

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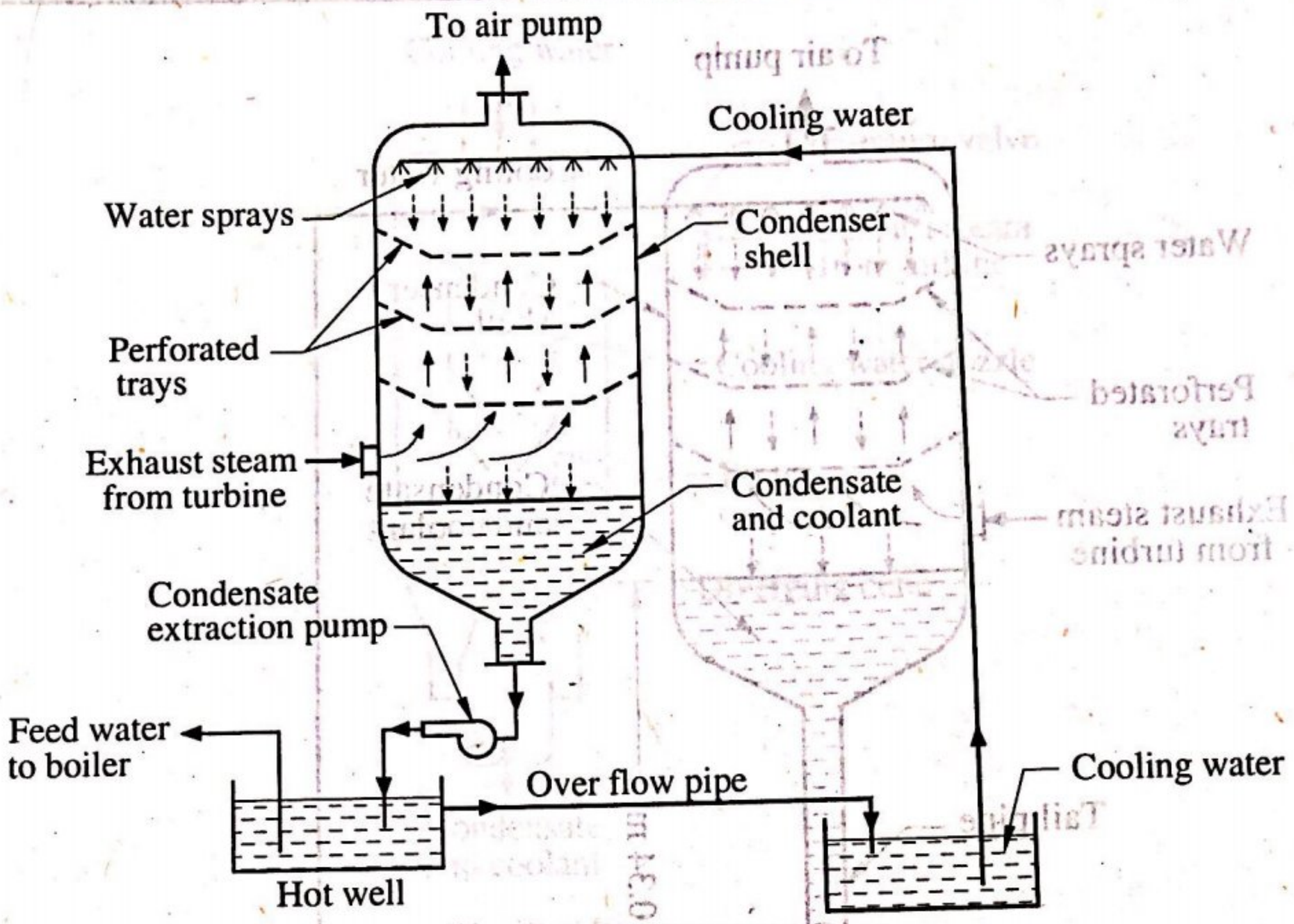


Fig. 7.4 Low level counter flow jet condenser

The low level counter flow jet condenser is more efficient than parallel flow because the hottest steam comes in contact with the hottest water. Therefore as steam goes more and more up, it comes into contact with more and more colder condensate and that enhances the steam condensation.

7.3.3 High level jet condenser

The arrangement of high level jet condenser is shown in Fig. 7.5. This type of condenser is also called ***barometric jet condenser*** since it is placed above the atmospheric pressure equivalent to 10.33 m of water.

High level jet condenser is provided at a high level with a long vertical tail pipe of length 10.34 m above the hot well. The exhaust steam enters at the bottom, flows upwards and meets the downcoming cooling water in the same way as that of low level jet condenser. The condensate and cooling water mixture is automatically drawn away by gravity without the use of any extraction pump due to existence of the barometric leg (tail pipe).

A separate pump is needed in this condenser for supplying cooling water to the condenser because of the height of the shell.

Advantage of this condenser is that the water from the hot well will not be able to rise into the condenser and flood the turbine. In a low level jet condenser, if the condensate extraction pump fails, the turbine will be flooded with water from the hotwell.

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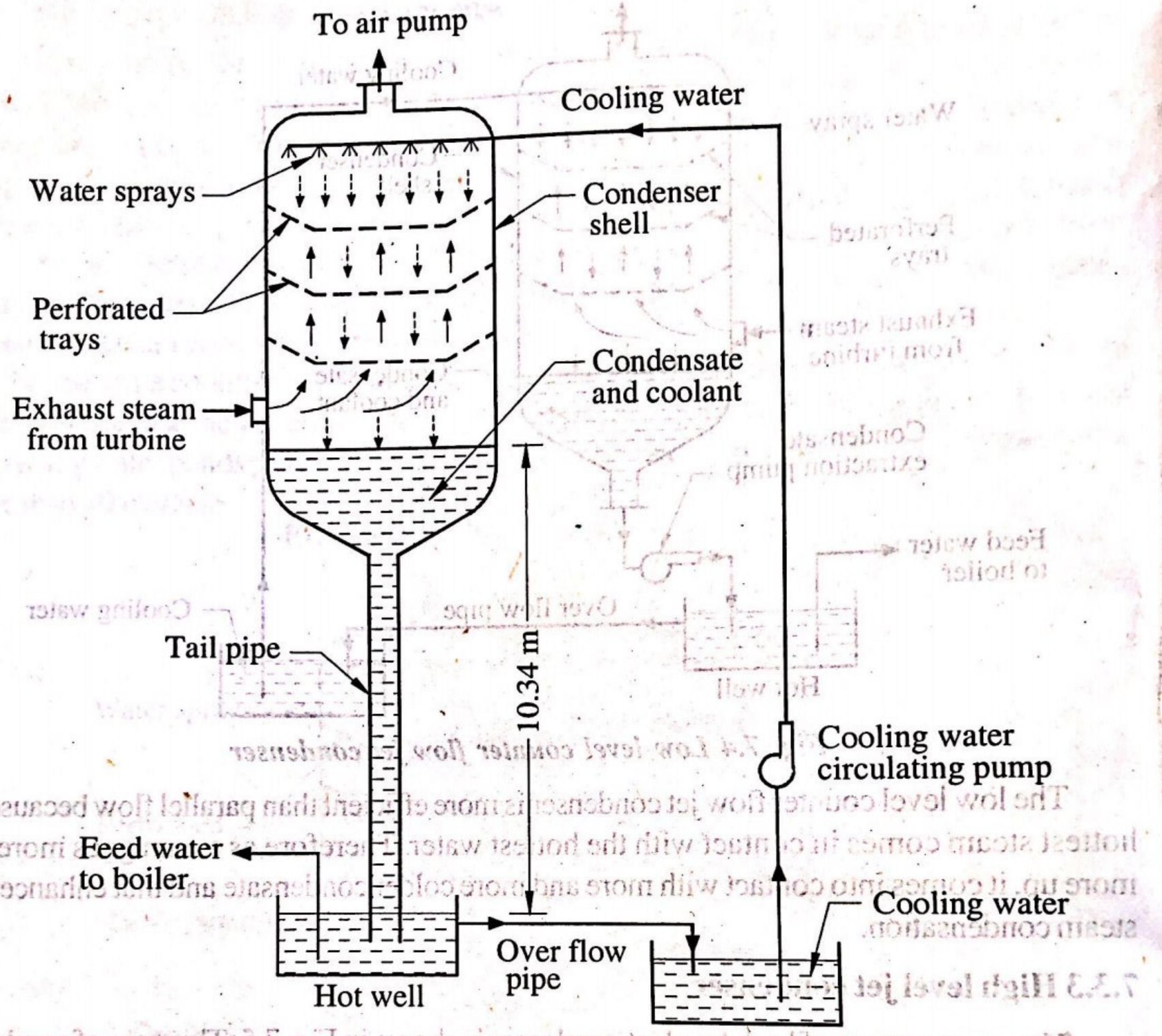


Fig. 7.5 High level jet condenser

7.3.4 Ejector condenser

Ejector condenser is a jet type condenser in which mixing of cooling water and steam takes place in a series of combining cones and the kinetic energy of the steam is expended to drain off the condensate and cooling water from the condenser. The arrangement of ejector condenser is shown in Fig. 7.6.

In this condenser, cooling water is forced through a series of cones and gets mixed with steam coming through ports. As the cooling water flows through the series of nozzles it suffers more and more pressure drop and at the same time its velocity gradually increases. Due to this pressure drop, more and more steam is drawn through the ports, gets intimately mixed with the cooling water jet and condenses thereafter.

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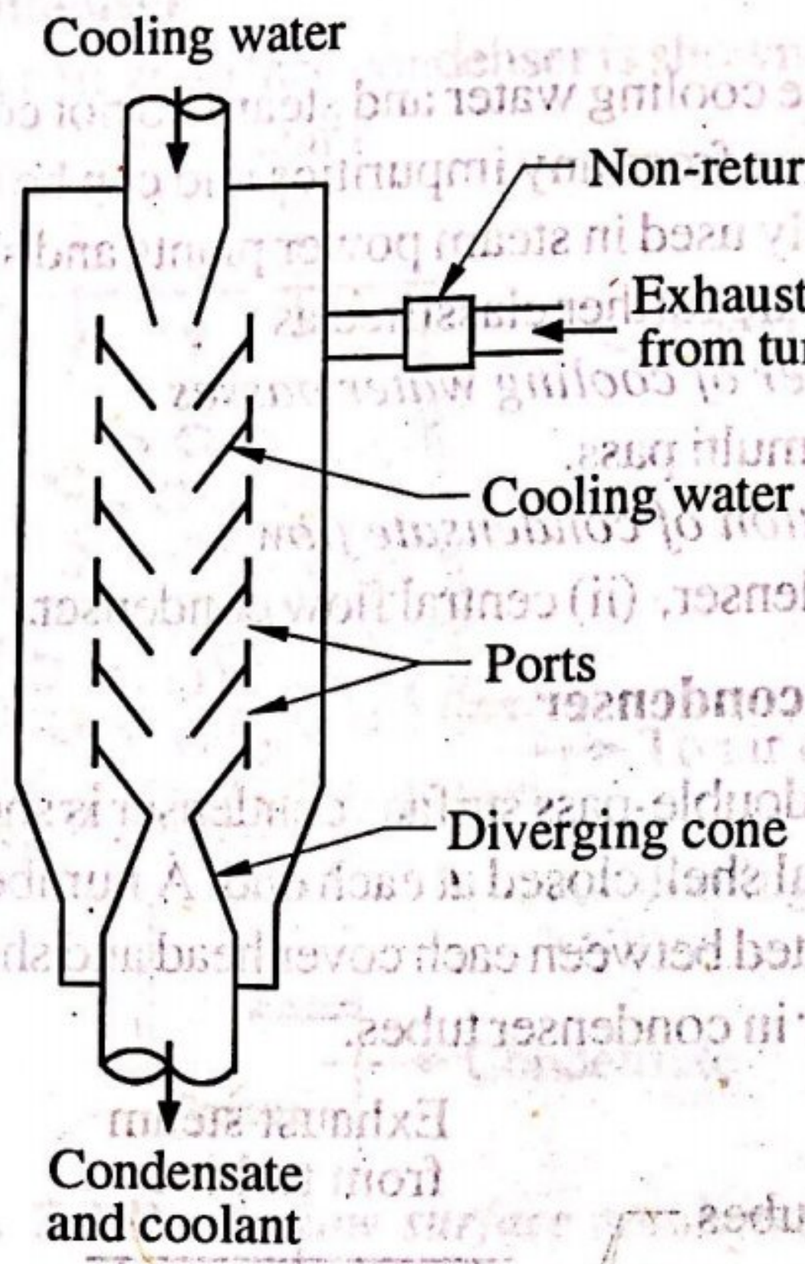


Fig. 7.6 Ejector condenser

A non return valve is provided on the steam inlet line to prevent the backflow of condensate in case of failure of the cooling water supply.

7.3.5 Advantages and disadvantages of jet condensers

- **Advantages :**
  (1) The quantity of cooling water required per kg of steam condensed is lower than that required for surface condenser.
  (2) They are simple in design and low in capital cost.
  (3) They require less floor space.
  (4) Lower maintenance cost.
  (5) Heat transfer from steam to water is more effective due to direct contact.
- **Disadvantages :**
  (1) The condensate cannot be directly used as feed water if cooling water is not pure.
  (2) The power required to run the air pump is higher.
  (3) If condensate is to be used, the cooling water should be clean and pure so that it can be used directly in the boiler.
  (4) Vacuum in jet condenser seldom exceeds 650 mm of Hg due to liberation of dissolved gases in the cooling water.
  (5) The system requires large lengths of piping, therefore, the piping cost is high.

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7.4 Surface condensers

In surface condenser, the cooling water and steam do not come in direct contact so that the condensate obtained is free from any impurities and can be used directly as feed water. Surface condensers are largely used in steam power plants and ships.

The surface condensers are further classified as :

(a) *According to number of cooling water passes :*

(i) single pass, (ii) multi pass.

(b) *According to direction of condensate flow :*

(i) down flow condenser, (ii) central flow condenser.

7.4.1 Double pass surface condenser

The arrangement of the double-pass surface condenser is shown in Fig. 7.7. It consists of a cast iron air-tight cylindrical shell closed at each end. A number of water tubes are fixed in the tube plates which are located between each cover head and shell. Circulating pump is used to circulate the cooling water in condenser tubes.

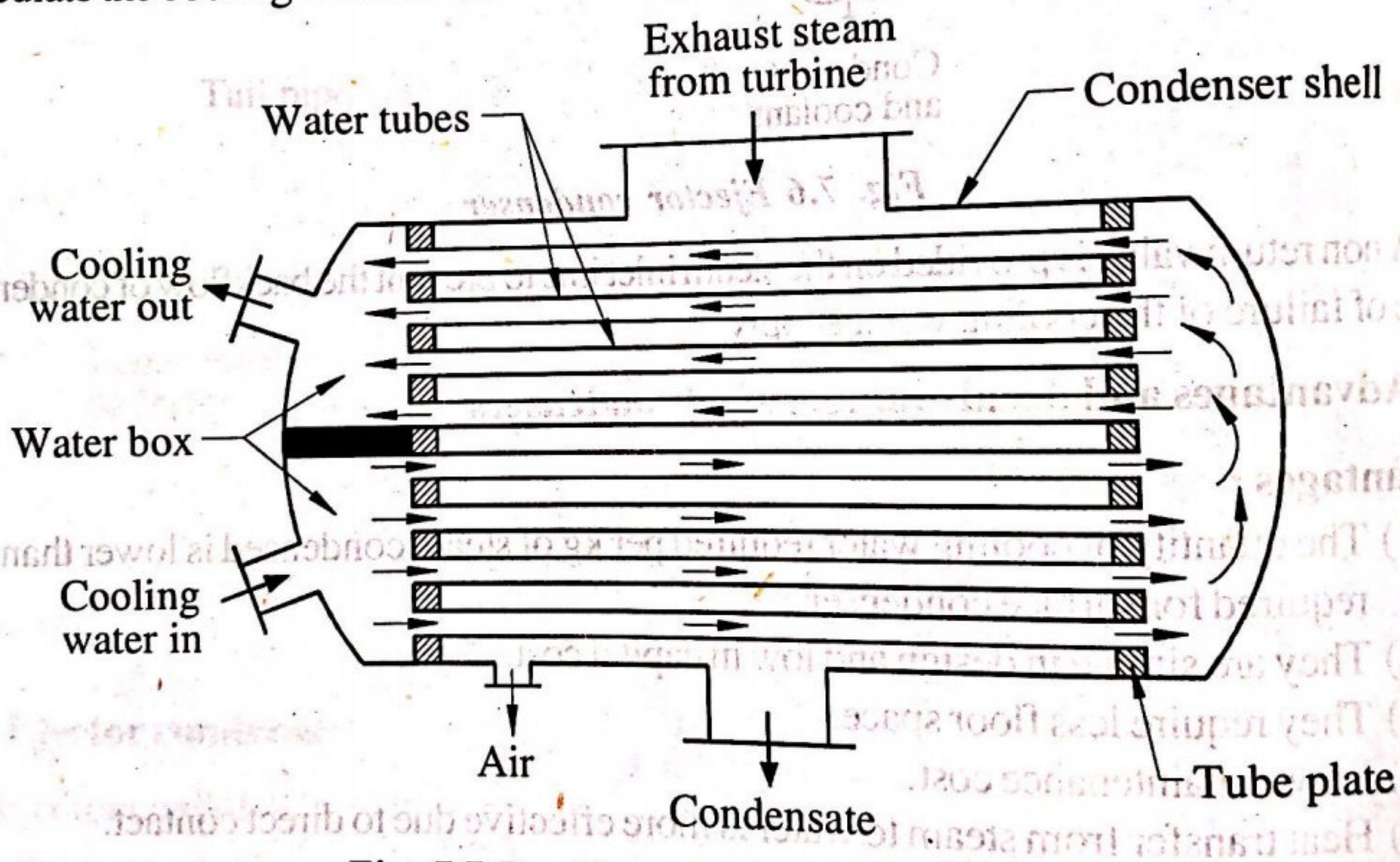


Fig. 7.7 Double pass surface condenser

Cooling water enters from bottom, flows through tubes, traverses back and leaves at the top, thus flowing through the entire length of the condenser twice.

Exhaust steam enters at the top and flows downward outside the tubes, gets condensed and is constantly being removed by condensate pump.

Air is continuously removed from a separate section by air pump. The air can also be removed by steam jet air ejectors instead of air pump.

These condensers may be of down flow or centre flow type.

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7.4.2 Down flow surface condenser

The cross-section of down flow surface condenser is shown in Fig. 7.8.

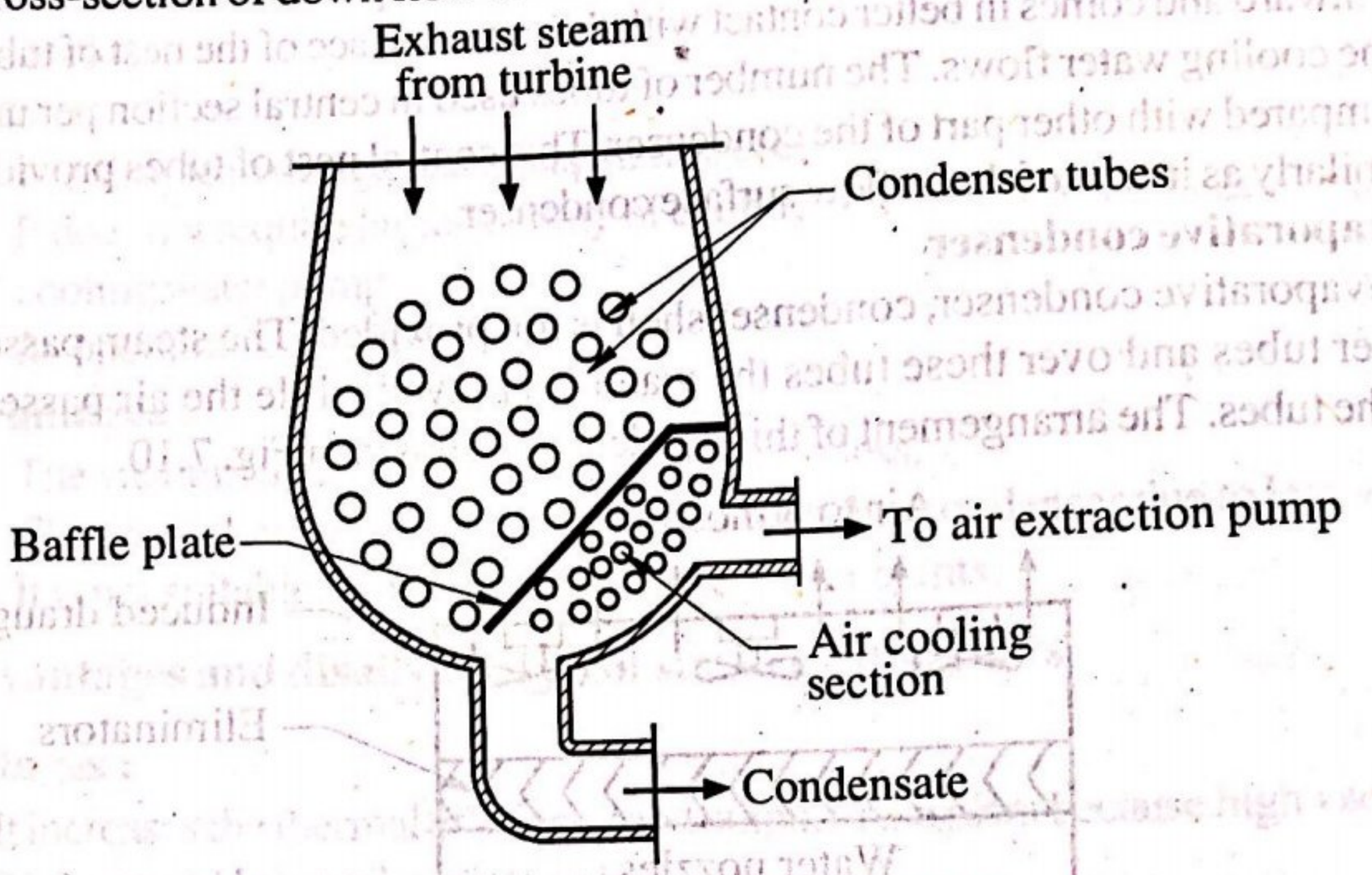


Fig. 7.8 Down flow surface condenser

The working principle of this condenser is similar to double pass surface condenser. A section of tubes near the air pump suction is screened off by providing a baffle. The number of tubes used in this section per unit area are more compared with other part of the condenser. The extensive cooling of air in this section increases the density of air going out and reduces the capacity of air pump required by as much as 50%.

7.4.3 Central flow surface condenser

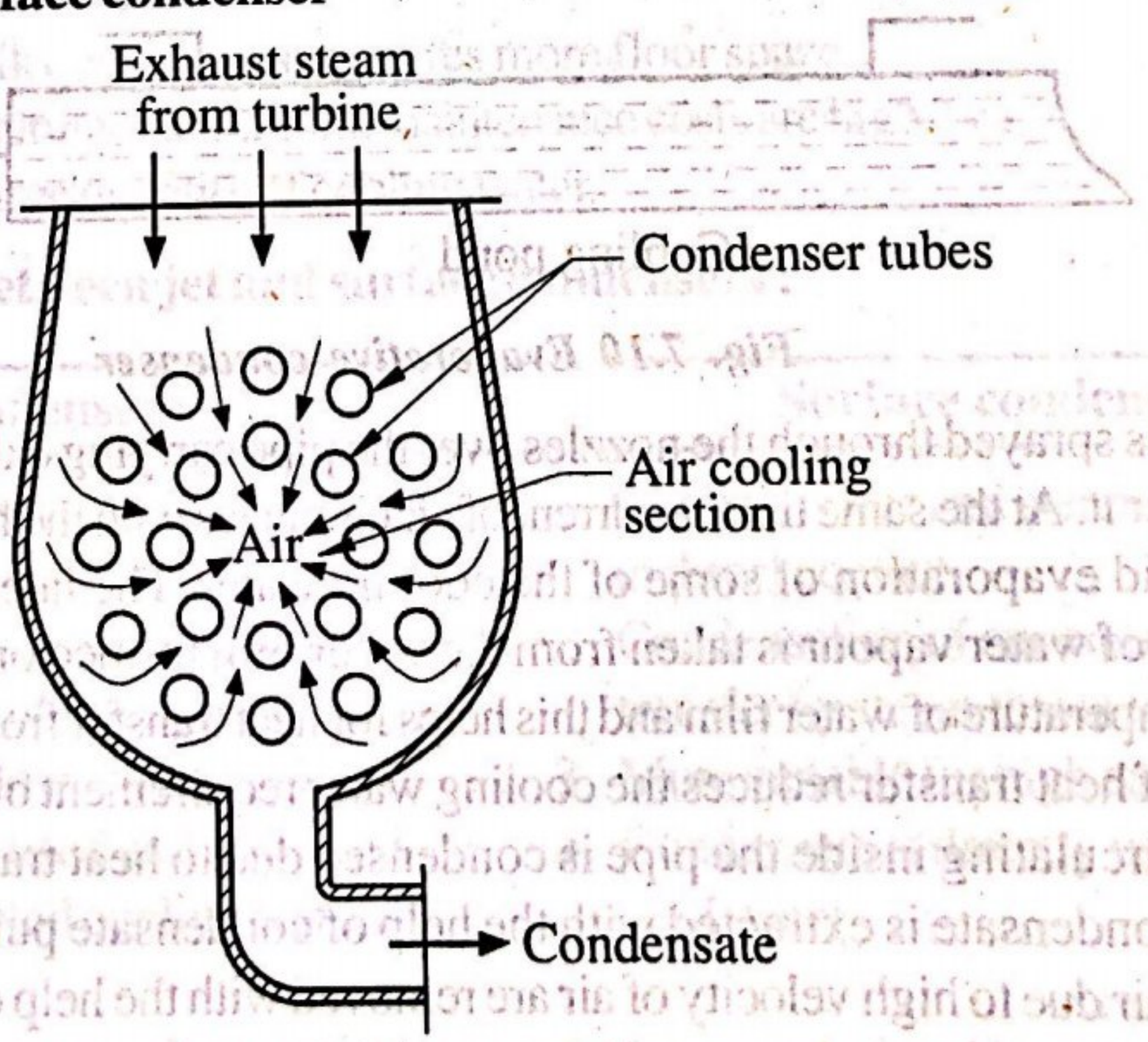


Fig. 7.9 Central flow condenser

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The cross-section of central flow surface condenser is shown in Fig. 7.9. The centre of this condenser is located the suction of an air extraction pump so that the entire steam moves radially inward and comes in better contact with the outer surface of the nest of tubes through which the cooling water flows. The number of tubes used in central section per unit area are more compared with other part of the condenser. This central nest of tubes provides cooling of air similarly as incase of down flow surface condenser.

7.4.4 Evaporative condenser

In evaporative condenser, condenser shell is not provided. The steam passes through condenser tubes and over these tubes the water is sprayed while the air passes upwards outside the tubes. The arrangement of this condenser is shown in Fig. 7.10.

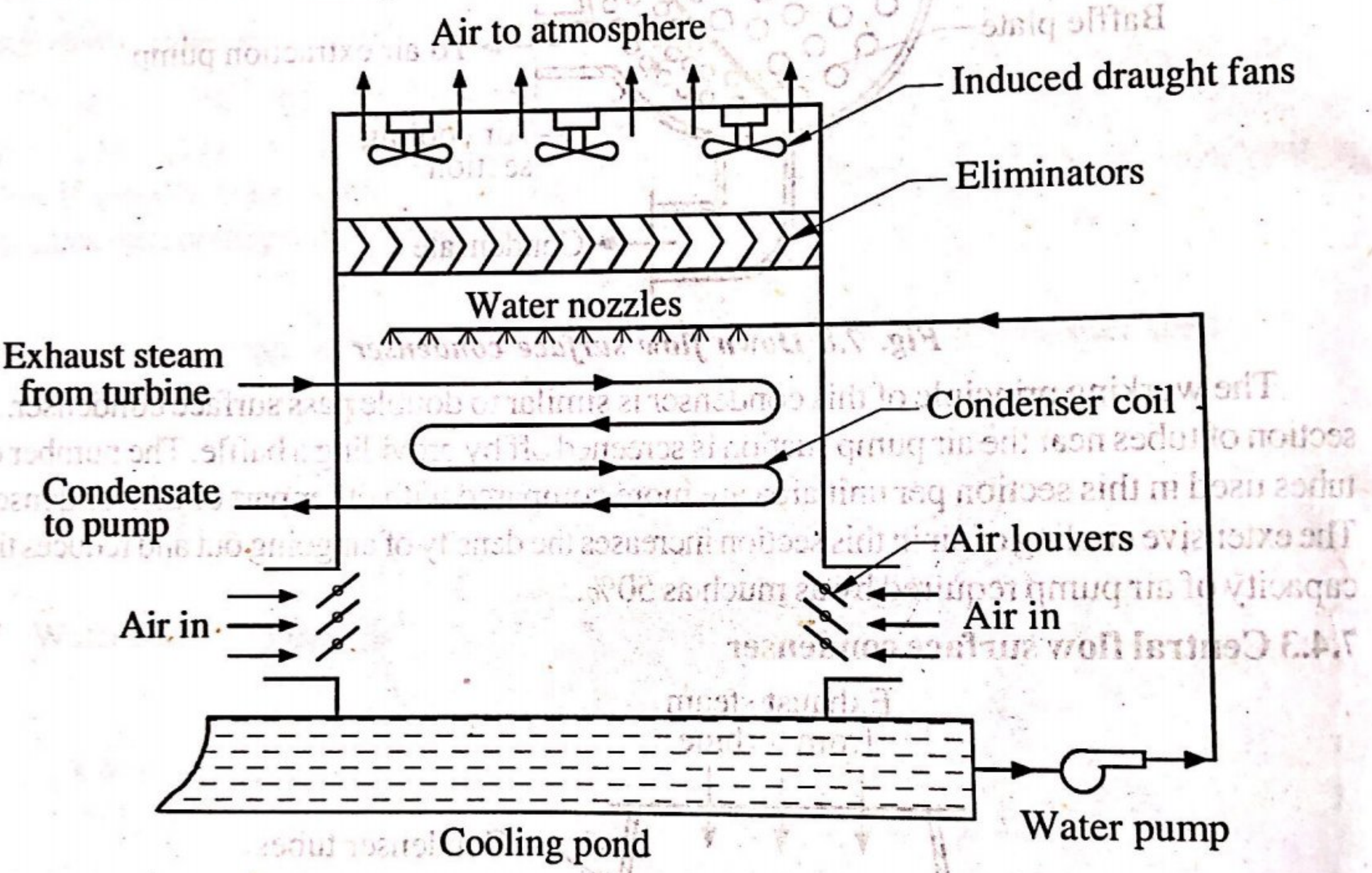


Fig. 7.10 Evaporative condenser

Water is sprayed through the nozzles over the pipe carrying exhaust steam and forms a thin film over it. At the same time, a current of air circulates with the help of induced draft fan, causing rapid evaporation of some of the cooling water. The latent heat required for the evaporation of water vapour is taken from the water film formed on the condenser coil and drops the temperature of water film and this helps for heat transfer from the steam to the water. This mode of heat transfer reduces the cooling water requirement of the condenser by 10%. The steam circulating inside the pipe is condensed due to heat transfer from steam to the water. The condensate is extracted with the help of condensate pump. The water particles carried with air due to high velocity of air are removed with the help of eliminator.

This condenser needs very small amount of circulating water and the make up water

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since the coolant is used again and again. These type of condensers are more preferable where acute shortage of cooling water exists. However these condensers are only suitable for small power plant.

- **Advantages :**
  (1) It is simple in design and cheap in first cost.
  (2) It does not require large quantity of cooling water, therefore, needs a small capacity cooling water pump.
  (3) Its operating cost is less.
- **Disadvantages :**
  (1) The vacuum maintained in this condenser is less.
  (2) The work done per kg of steam is less with this condenser due to less vacuum.
  (3) It is not suitable for medium and large power plants.

7.4.5 Advantages and disadvantages of surface condensers

- **Advantages :**
  (1) It increases the thermal efficiency and output of the plant because high vacuum (about 73.5 cm of Hg) can be obtained.
  (2) The condensate can be reused as boiler feed-water.
  (3) Auxiliary power requirement is less than that of a jet condenser.
  (4) Less amount of air is carried to the boiler.
  (5) Any kind of feed water can be used.
  (6) It can develop high vacuum, therefore, these are suitable for large capacity power plants.
- **Disadvantages :**
  (1) It is very bulky and as such requires more floor space.
  (2) Its manufacturing, running and maintenance costs are high.
  (3) It requires large quantity of cooling water.

7.4.6 Comparison between jet and surface condensers : [Dec. '13]

Jet condensers	Surface condensers
1. Cooling water and steam come in direct contact.	1. Cooling water and steam do not come in direct contact.
2. Condensation of steam is due to heat transfer by mixing of steam with water.	2. Condensation of steam is due to heat transfer by steam to water through tubes.
3. More suitable for low capacity plants.	3. More suitable for high capacity plants.
4. Condensate is not used as boiler feed water until the treated cooling water is used as coolant.	4. Condensate is directly used as boiler feed water.
5. Vacuum created is upto 600 mm of Hg.	5. Vacuum created is upto 730 mm of Hg.

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6. Its maintenance cost is low.	6. Its maintenance cost is high.
7. The condensing plant is economical and simple.	7. The condensing plant is costly and complicated.
8. More power is required for air pump.	8. Less power is required for air pump.

7.4.7 Requirements of a good surface condenser

(1) There should be uniform distribution of exhaust steam throughout the heat transfer surface of the condenser.
(2) There should not be sub-cooling of condensate.
(3) There should not be any leakage of air into the condenser.
(4) There should not be any tube leakage.
(5) The heat transfer surface in contact with cooling water must be free from any deposit as scaling reduces the efficiency of heat exchangers.

7.5 Sources of air in condenser

The sources of air in condenser are as follows :
(1) The boiler feed water contains dissolved air. From the boiler it is carried off by steam and to the turbine and finally to the condenser.
(2) The air leaks through the joints, packings and glands into the condenser where the pressure is below the atmospheric pressure. This can be reduced by proper workmanship while making the joints.
(3) Air dissolved with cooling water is carried in case of jet condensers.

Normally the quantity of air leakage in surface condenser is 0.05% of steam condensed.

7.6 Effects of air leakage

The effects of air leakage in a condenser are given below :
(1) It increases the back pressure on the turbine with the effect that there is less enthalpy drop and ***low thermal efficiency of the plant.***
(2) The pressure of air in the condenser lowers the partial pressure of steam which means steam will condense at a lower temperature and that will require ***greater amount of cooling water.***
(3) It reduces the rate of condensation of steam, because air having poor thermal conductivity ***reduces the overall heat transfer from the steam air mixture.***
(4) The presence of air in the condenser ***increases the corrosive action.***

7.7 Methods of obtaining maximum vacuum in condenser [Dec. '13]

The methods used to obtain maximum vacuum in condensers are discussed in following:
(1) Air pump : Air pump are used to extract the air and other non-condensable gases from the condenser to maintain a desired vacuum. They are usually classified as
(a) *Wet air pump :* It removes a mixture of condensate and air with other non-condensable gases.

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(b) *Dry air pump :* It removes the air and other non-condensable gases only.

It is becoming increasingly common to have separate pumps for extracting the air and water from a condenser.

(2) Steam jet air ejector : It is used to remove air from condenser when a wet pump (also called extraction pump) is employed. The working principle of steam jet air ejector is shown in Fig. 7.11. It consists of convergent-divergent nozzle and a diffuser.

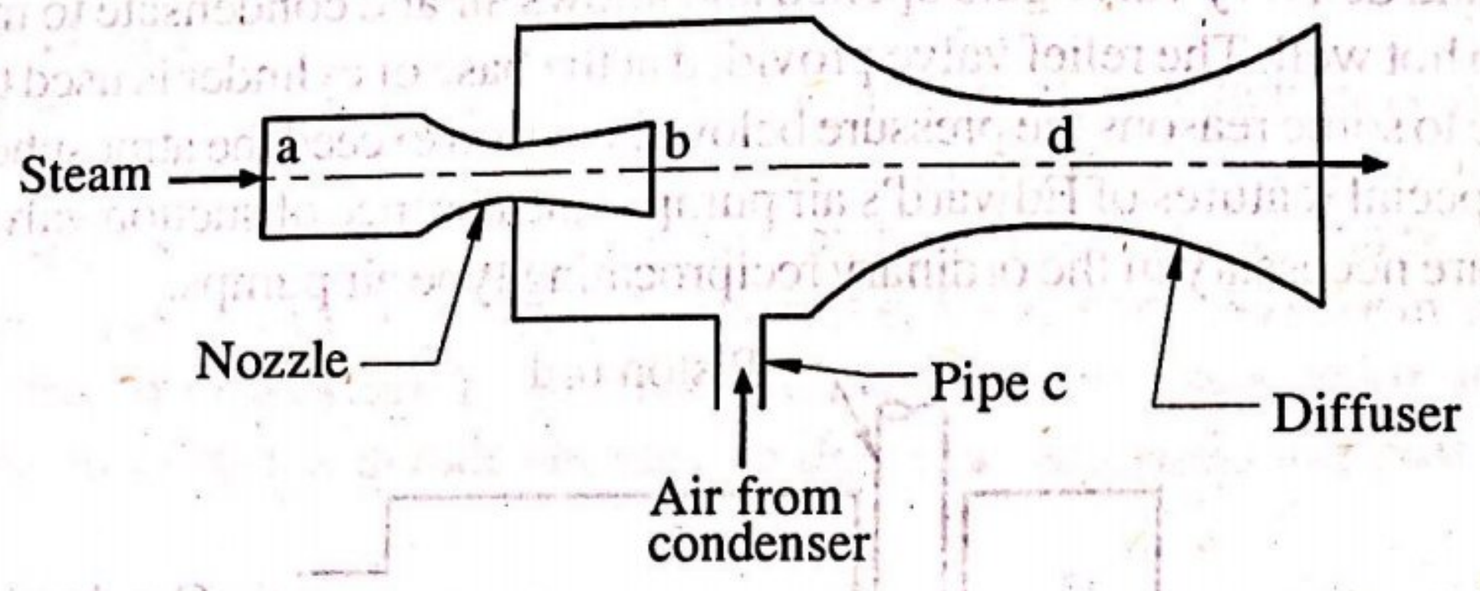


Fig. 7.11 Steam jet air ejector

The steam at a high pressure enters at '*a*' and feeds directly the nozzle. The steam expands in the nozzle and leaves the nozzle at '*b*' with high velocity and a small pressure depending on the vacuum in the condenser. Pipe '*c*' is connected to the condenser from where the air along with small amount of vapour forces it to mix with the low pressure steam at '*b*'. The mixture of steam and air moves forward into the diffuser '*d*' where the kinetic energy of the mixture is converted into pressure energy and it leaves the diffuser.

In the case of ejectors used for steam power plants where a high vacuum pressure is maintained in the condenser, it is necessary to use two, or perhaps three ejectors in series to obtain maximum vacuum.

The steam jet air ejectors are simple in construction, cheap, highly efficient and no moving parts in it.

(3) De-aerated feed water : The deaeration is the process of removal of non condensable gases from feed water. It helps both in maintaining better vacuum in the condenser and controlling corrosion of the steel shell and piping of the steam power plant.

(4) Air tight joints : The air leaks through the joints, packings and glands into the condenser. This can be reduced by proper workmanship while making the joints and these are maintained as such by proper inspection from time to time.

7.8 Edward air pump

The Edward air pump is used to remove the air and condensate from the condenser. It is a ***wet air pump*** of the reciprocating type.

The Edward's air pump consists of a piston which has a conical head as shown in Fig. 7.12. The bottom of the pump casing is also of conical shape in order that piston head can be easily seated. The piston slides inside the barrel having a cover which has a number of delivery

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valves. On the down stroke of the piston a partial vacuum is produced above it since delivery valves are closed and sealed by water. When the piston uncovers the port 'A' air and condensate from the condenser rush into the space above the piston. As the piston further moves down the conical part displaces the condensate which has collected in the bottom portion of the pump during upward stroke of piston and forces it to flow into the upper portion through the ports 'B'. Now when the piston moves upward it raises the pressure slightly over that of the atmospheric and delivery valve gets opened and allows air and condensate to move out and flow finally to hot well. The relief valve provided at the base of cylinder is used to release the pressure if due to some reasons the pressure below the piston exceed the atmospheric pressure.

The special features of Edward's air pump is the absence of suction valve and bucket valve, which are necessary in the ordinary reciprocating type air pumps.

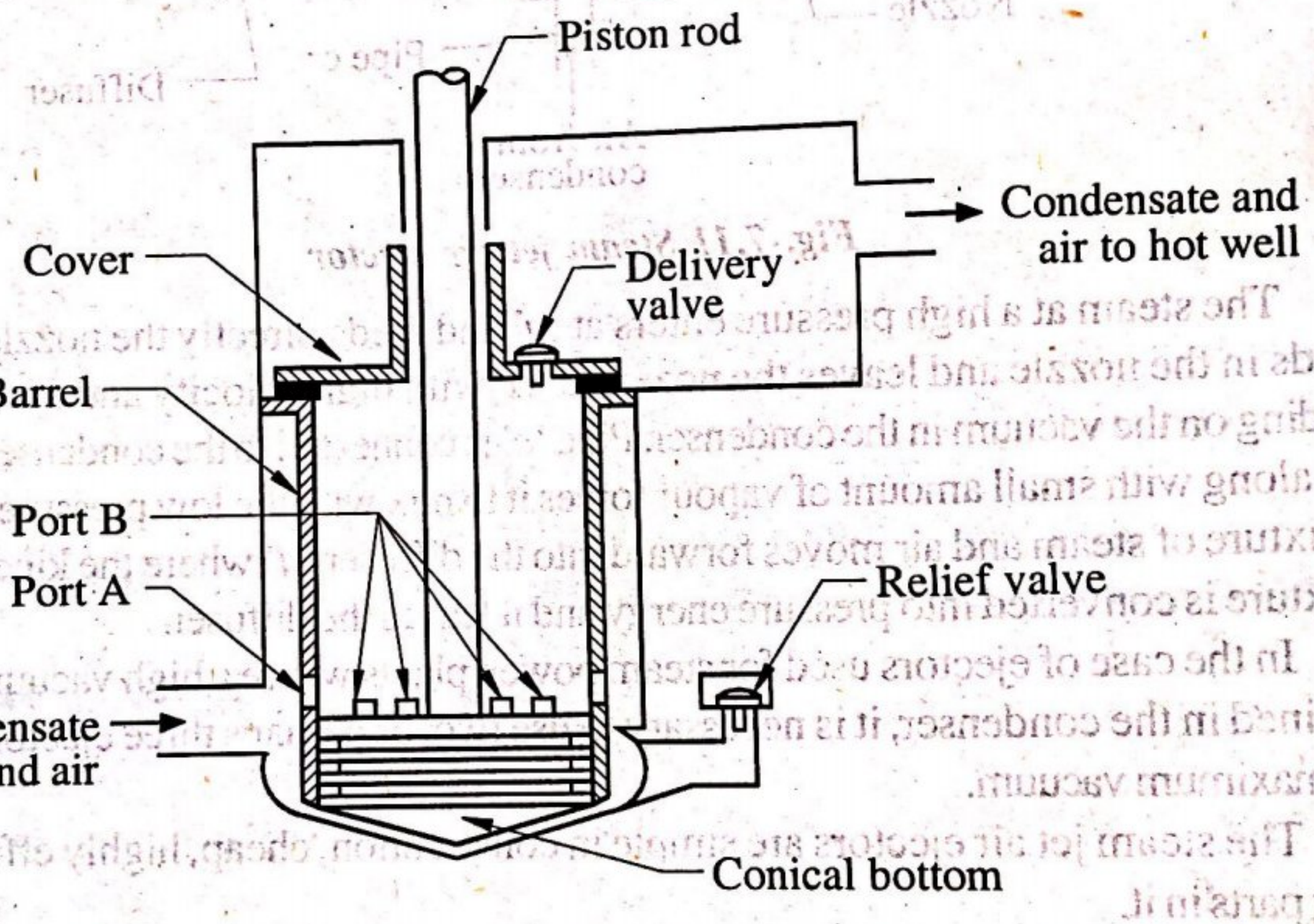


Fig. 7.12 Edward air pump

7.9 Dalton's law of partial pressure

It states "*The pressure of the mixture of air and steam is equal to the sum of the pressures which each constituent would exert, if it occupied the same space by itself.*" Mathematically, pressure in the condenser containing mixture of air and steam,

$$p_c = p_a + p_s$$

where, p_c = pressure in condenser,

p_a = partial pressure of air, and

p_s = partial pressure of steam.

In most of the cases, we are required to find partial pressure of air, therefore Dalton's

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law may also be used as :

$$p_a = p_c - p_s$$

***Note** : For more information about partial pressure refer book 'Engineering Thermodynamics' by the same authors.*

7.10 Vacuum and condenser efficiency

- **Vacuum efficiency :**

The minimum absolute pressure at the steam inlet of a condenser is the saturation pressure corresponding to the temperature of the condensed steam. The corresponding vacuum is called ***ideal vacuum***. The ideal vacuum is the maximum vacuum that can be obtained in a condensing plant, with no air present at that temperature. The ***actual pressure*** in the condenser is greater than the ideal pressure by an amount equal to the pressure of air present in the condenser. *The ratio of the actual vacuum to the ideal vacuum is known as* ***vacuum efficiency***.

Ideal vacuum possible without air leakage = $(p_b - p_s)$

Actual vacuum existing in condenser due to air leakage

$$= p_b - p_c = p_b - (p_a + p_s)$$

$$\therefore \text{Vacuum efficiency, } \eta_v = \frac{p_b - (p_a + p_s)}{p_b - p_s} = \frac{p_b - p_c}{p_b - p_s}$$

where p_s = saturation pressure of steam corresponding to the steam temperature entering into the condenser.

p_c = Total pressure of the air and steam in the condenser $(p_a + p_s)$.

p_b = atmospheric or barometric pressure.

The leakage of air and insufficient quantity of cooling water decreases the vacuum efficiency of the condenser.

- **Condenser efficiency :**

The condenser efficiency is defined as the ratio of actual rise in the temperature of cooling water to the maximum possible rise in temperature of cooling water.
Mathematically,

$$\eta_c = \frac{\text{Actual rise in the temperature of cooling water}}{\text{Inlet temperature of steam} - \text{Inlet temperature of cooling water}}$$

$$\therefore \eta_c = \frac{T_o - T_i}{T_s - T_i}$$

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where T_o = Outlet temperature of cooling water,

T_i = Inlet temperature of cooling water,

T_s = Saturation temperature of steam corresponding to actual absolute pressure in the condenser.

7.11 Mass of cooling water required for condensation of steam

The mass of cooling water required for condenser to condense a given amount of steam is calculated as follows :

Let m_w = Mass of cooling water,

m_s = Mass of steam condensed (i.e. condensate),

h = Total heat of steam entering the condenser,

h_{f_1} = Total heat in condensate,

T_i = Inlet temperature of circulating water, and

T_o = Outlet temperature of circulating water.

We know that heat lost by steam

$$= m_s (h - h_{f_1})$$

Heat gained by cooling water

$$= m_w c_w (T_o - T_i)$$

From energy balance, heat gained by cooling water = Heat lost by steam

$$\therefore m_w c_w (T_o - T_i) = m_s (h - h_{f_1})$$

$$\therefore \boxed{m_w = \frac{m_s (h - h_{f_1})}{c_w (T_o - T_i)}}$$

Problem 7.1 *Steam enters the condenser at 36° C. The condenser vacuum is 70 cm of mercury when the barometer reads 75.5 cm of Hg. Determine the vacuum efficiency. Estimate the mass of air present in the condenser per kg of steam.*

Solution : Given data

$T_s = 36^o C$, condenser vacuum = 70 cm of Hg = 0.7 m of Hg.

Barometer reading = 75.5 cm of Hg = 0.755 m of Hg., $\eta_v = ?$, $m_a = ?$

From steam tables, saturation pressure of steam at $36^o C$,

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$p_s = 0.0595$ bar, and $v_g = 23.94$ m^3/kg.

Standard atmospheric pressure = 76 cm of Hg = 1.013 bar.

$$\therefore\ p_s = 0.0595 \text{ bar} = \frac{0.0595\times10^5}{13.6\times1000\times9.81} = 0.04459 \text{ m of Hg.} = 4.459 \text{ cm of Hg.}$$

Absolute pressure in condenser, p_c ;

p_c = Barometer reading – condenser vacuum

= 75.5 – 70 = 5.5 cm of Hg.

Vacuum corrected to the standard barometer reading

= Standard barometer reading – Absolute pressure in condenser

= 76 – 5.5 = 70.5 cm of Hg.

We know that ideal vacuum

= Standard barometer reading – Ideal pressure (saturation pressure of steam)

= 76 – 4.459 = 71.541 cm of Hg.

$$\therefore \text{ Vacuum efficiency, } \eta_v = \frac{\text{Actual vacuum}}{\text{Ideal vacuum}} = \frac{70.5}{71.541}$$

$$= 0.9854 = \mathbf{98.54\ \%} \qquad \textbf{Ans}$$

Partial pressure of air, $p_a = p_c - p_s$

$= 5.5 - 4.459 = 1.041$ cm of Hg

$= 1.041\times10^{-2}\times13.6\times1000\times9.81$ N/m^2

$= 0.01389$ bar

∴ Mass of air associated with 1 kg of steam;

$$m_a = \frac{p_a v}{RT} = \frac{0.01389\times10^5\times23.94}{287\times(273+36)}$$

$$= \mathbf{0.3749 \text{ kg of air/kg of steam.}} \qquad \textbf{Ans}$$

Problem 7.2 *Exhaust steam having a quality of 0.9 enter a surface condenser at an absolute pressure of 0.13 bar and comes out as a water at 45° C. The circulating water enters at 30° C and leaves at 40° C. Estimate quantity of circulating water and condenser efficiency.* ***[Dec. '10, 7 marks]***

Solution : Given data

$x = 0.9,\ p_s = 0.13$ bar, $T_{f_1} = 45^\circ C$, $T_i = 30^\circ C$, $T_o = 40^\circ C$, $m_w / m_s = ?$, $\eta_c = ?$

From steam tables, at 0.13 bar saturation pressure

$T_s = 51^\circ C$, $h_f = 213.7$ kJ/kg, $h_{fg} = 2380.2$ kJ/kg,

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and at $45^{\circ}C$ temperature, $h_{f_1} = 188.4$ kJ/kg

Enthalpy of steam entering to the condenser,

$$h = h_f + xh_{fg} = 213.7 + 0.9 \times 2380.2$$
$$= 2355.88 \text{ kJ/kg.}$$

By energy balance,

$$m_w c_w (T_o - T_i) = m_s (h - h_{f_1})$$

$$\therefore \frac{m_w}{m_s} = \frac{h - h_{f_1}}{c_w (T_o - T_i)} = \frac{2355.88 - 188.4}{4.182(40 - 30)}$$

$$= \mathbf{51.828 \text{ kg of water/kg of steam.}} \quad \textit{Ans}$$

$$\text{Condenser efficiency, } \eta_c = \frac{T_o - T_i}{T_s - T_i} = \frac{40 - 30}{51 - 30}$$

$$= 0.4762 = \mathbf{47.62\%} \quad \textit{Ans}$$

Problem 7.3 *The following readings were taken during a test on a surface condenser : Mean condenser temperature = $36^{\circ}C$, Hot well temperature = $30^{\circ}C$, condenser vacuum = 70 cm of Hg, barometer reading = 76 cm of Hg, Condensate collected 12 kg/min, cooling water enters at $23^{\circ}C$ and leaves at $32^{\circ}C$, flow rate being 38,000 kg/h. Calculate (i) mass of air present per cubic metre of condenser, (ii) quality of steam at condenser inlet, (iii) vacuum efficiency. and (iv) condenser efficiency.*

Solution : Given data ***[same as Jan. '13 and June '11, 7 marks]***

$T_s = 36^{\circ}C$, $T_h = 30^{\circ}C$, condenser vacuum = 70 cm of Hg, Barometer reading = 76 cm of Hg, $m_s = 12$ kg/min $= 12 \times 60$ kg/h, $T_i = 23^{\circ}C$, $T_o = 32^{\circ}C$, $m_w = 38{,}000$ kg/h, m_a = ?, x = ?, η_v = ?, η_c = ?

(i) *Mass of air present per cubic metre of condenser* ($v = 1 m^3$),

From steam tables at $36^{\circ}C$, $p_s = 0.0595$ bar

Absolute pressure inside the condenser, p_c

= Barometer reading – condenser vacuum

$= 76 - 70 = 6$ cm of Hg

$= 6 \times 10^{-2} \times 13.6 \times 1000 \times 9.81 = 8004.96 \text{ N/m}^2$

$\cong 0.08$ bar

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$\therefore$ Partial pressure of air, $p_a = p_c - p_s$

$= 0.08 - 0.0595 = 0.0205$ bar

$\therefore$ Mass of air present, $m_a = \dfrac{p_a v}{RT} = \dfrac{0.0205 \times 10^5 \times 1}{287 \times (273+36)}$

$= \mathbf{0.0231\ kg/m^3}$ ***Ans***

(ii) Quality of steam at condenser inlet,

Let x = Dryness fraction (quality) of steam at condenser inlet.

From steam tables, corresponding to temperature of $36^\circ C$.

$h_f = 150.9$ kJ/kg, $h_{fg} = 2416.2$ kJ/kg,

and corresponding to a hot-well temperature of $30^\circ C$

$h_{f_1} = 125.8$ kJ/kg

We know that total heat of entering steam,

$h = h_f + x h_{fg} = 150.9 + x \times 2416.2$

We also know that mass of cooling water (m_w),

$\therefore\ m_w = \dfrac{m_s (h - h_{f_1})}{c_w (T_o - T_i)}$

$\therefore 38000 = \dfrac{12 \times 60 [150.9 + x \times 2416.2 - 125.8]}{4.18(32-23)}$

$\therefore\ x = \mathbf{0.812}$ ***Ans***

(iii) Vacuum efficiency

We know that corresponding to a condensation temperature of $36^\circ C$,

Ideal pressure of steam = 0.0595 bar $= \dfrac{0.0595 \times 10^5}{13.6 \times 1000 \times 9.81}$

$= 0.0446$ m of Hg $= 4.46$ cm of Hg

$\therefore$ Ideal vacuum = Barometer reading – Ideal pressure

$= 76 - 4.46 = 71.54$ cm of Hg

$\therefore$ Vacuum efficiency, $\eta_v = \dfrac{\text{Actual vacuum}}{\text{Ideal vacuum}} = \dfrac{70}{71.54} = 0.9784$

$= \mathbf{97.84\ \%}$ ***Ans***

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(iv) Condenser efficiency,

$$\eta_c = \frac{T_o - T_i}{T_s - T_i} = \frac{32-23}{36-23} = 0.6923 = \mathbf{69.23\%}$$ **Ans**

Problem 7.4 *A surface condenser deals with 13600 kg of steam per hour at 0.09 bar pressure. The steam is 0.85 dry at condensate temperature. The air extraction pipe is at $36^\circ C$. The air leakage amounts to 7.26 kg/hr. Determine (i) The surface required, if heat transmission rate is 3.97 kJ/cm^2 per second. (ii) Cylinder diameter of pump for dry air, if it is to be single acting at 60 rpm with stroke to bore ratio of 1.25 and volumetric efficiency of 0.85.* **[G.U., January '08, 8 Marks]**

Solution : Given data

$m_s = 13600$ kg/h, $p_c = 0.09$ bar, $x = 0.85$, $T_a = 36^\circ C$, $m_a = 7.26$ kg/h,

Q = 3.97 kJ/cm^2 per second, N = 60 rpm, L/D = 1.25, $\eta_{vol} = 0.85$, surface area = ?,

D = ?

From steam tables, at 0.09 bar pressure

$h_f = 183.3$ kJ/kg, $h_{fg} = 2397.7$ kJ/kg

and heat of condensate at $36^\circ C$,

$h_{f_1} = 150.9$ kJ/kg,

We know that total heat of entering steam,

$$h = h_f + xh_{fg} = 183.3 + 0.85 \times 2397.7 = 2221.345 \text{ kJ/kg}$$

Heat extracted per second from steam $= m_s(h - h_{f_1})$

$$= \frac{13600}{3600}[2221.345 - 150.9]$$

$$= 7821.68 \text{ kJ/s}$$

(i) The surface required in cm^2

$$= \frac{\text{Heat extracted per second from steam}}{\text{Heat transfer rate in kJ/s per cm}^2}$$

$$= \frac{7821.68}{3.97} = \mathbf{1970.196 \text{ cm}^2}$$ **Ans**

(ii) Cylinder diameter of pump,

From steam tables, partial pressure of steam at $36^\circ C$

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$$p_s = 0.0595 \text{ bar}$$

$\therefore$ Partial pressure of air, $p_a = p_c - p_s = 0.09 - 0.0595$

$= 0.0305$ bar

Using characteristic gas equation,

$$p_a V = m_a RT$$

$$\therefore 0.0305 \times 10^5 \times V = \frac{7.26}{3600} \times 287 \times (273+36)$$

$$\therefore V = 0.0586\, m^3/s$$

$$\text{Capacity of air pump/s} = \frac{V \text{ in } m^3/s}{\eta_{vol}} = \frac{0.0586}{0.85} = 0.0689\ m^3/s$$

$$\text{Capacity of the pump per stroke} = \frac{\text{Capacity in } m^3/s}{rpm/60} = \frac{0.0689}{60/60} = 0.0689\ \text{m}^3$$

Also, $\frac{\pi}{4} \times d^2 \times L = 0.0689$

$$\therefore \frac{\pi}{4} \times d^2 \times 1.25 \times d = 0.0689$$

$$\therefore d^3 = \frac{0.0689 \times 4}{\pi \times 1.25} = 0.07018$$

$\therefore$ $d = 0.4125$ m $=$ **41.25 cm** **Ans**

Problem 7.5 *During a trial on a steam condenser, the following observations were recorded.*

Condenser vacuum	*680 mm hg*
Barometer reading	*764 mm Hg*
Mean condenser temperature	*36.2°C*
Hot well temperature	*30°C*
condensate formed/hour	*1780 kg*
cooling water inlet temperature	*20°C*
cooling water outler temperature	*32°C*
Quantity of cooling water	*1250 kg/min*

Determine :

(i) Condenser vacuum corrected to standard barometer.

(ii) Vacuum efficiency.

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(iii) Under cooling of condensate.
(iv) Condition of steam at inlet of condenser.
(v) Condenser efficiency.
(vi) Mass of air present per kg of condensed steam. ***[G.U., Dec. '04, 8 Marks]***

Solution : Given data :

Condenser vacuum = 680 mm of Hg, Barometer reading = 764 mm of Hg,

$T_s = 36.2^\circ C$, $T_h = 30^\circ C$, $m_s = 1780$ kg/h, $T_i = 20^\circ C$, $T_o = 32^\circ C$, $m_w = 1250$ kg/min.

(i) Condenser vacuum corrected to standard barometer

= standard barometric pressure – (barometric pressure – gauge pressure)

= 760 – (764 – 680) = 676 mm of Hg.

But 1 mm of Hg = 1.333×10^{-3} bar

676 mm of Hg = $676 \times 1.333 \times 10^{-3}$ = **0.9011 bar** ***Ans***

(ii) Vacuum efficiency

From steam tables, at 36.2°C temperature

saturation pressure, $p_s = 0.06$ bar

Ideal vacuum = Barometer reading – Ideal pressure

$= 764 \times 1.333 \times 10^{-3} - 0.06$

$= 0.9584$ bar

$$\therefore \text{Vacuum efficiency, } \eta_v = \frac{\text{Actual vacuum}}{\text{Ideal vacuum}} = \frac{680 \times 1.333 \times 10^{-3}}{0.9584}$$

$= 0.9458 =$ **94.58 %** ***Ans***

(iii) Under cooling of condensate :

= condensate temperature – Hot well temperature

= 36.2 – 30 = **6.2°C** ***Ans***

(iv) Condition of steam at inlet of condenser

Absolute condenser pressure = Barometric pressure – condenser pressure

= 764 – 680 = 84 mm of Hg.

$= 84 \times 1.333 \times 10^{-3} = 0.1119$ bar

From steam tables, at 0.1119 bar pressure,

$h_f = 199.7$ kJ/kg, $h_{fg} = 2388.3$ kJ/kg

Enthalpy of condensate corresponding to hot well temperature of 30°C, $h_{f_1} = 125.8$ kJ/kg.

We know that total heat of entering steam,

$$h = h_f + x h_{fg} = 199.7 + x \times 2388.3$$

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We also know that mass of cooling water (m_w);

$$m_w = \frac{m_s(h - h_{f_1})}{c_w(T_o - T_i)}; \quad \frac{1250}{60} = \frac{\left(\frac{1780}{3600}\right)(199.7 + x \times 2388.3 - 125.8)}{4.186 \times (32 - 20)}$$

$\therefore$ $x =$ **0.855** ***Ans***

(v) Condenser efficiency :

From steam tables, saturation temperature corresponding to 0.1119 bar, $T_s = 47.7°C$

$\therefore$ Condenser efficiency, $\eta_c = \frac{T_o - T_i}{T_s - T_i} = \frac{32 - 20}{47.7 - 20} = 0.4332 =$ **43.32 %** ***Ans***

(vi) Mass of air present per kg of steam

From steam tables, at mean condensate temperature of 36.2°C,

specific volume of steam, $v_g = 23.74\ m^3/kg$

$\therefore$ Partial pressure of air, $p_a = p_c - p_s$

$= 0.1119 - 0.06 = 0.0519$ bar

$\therefore$ Mass of air present per kg of condensed steam,

$$m_a = \frac{p_a V}{RT} = \frac{0.0519 \times 10^5 \times 23.74}{287 \times (273 + 36.2)} = \mathbf{1.3884\ kg} \quad \textbf{\textit{Ans}}$$

Problem 7.6 *The following data collected from a steam turbine of 25 MW capacity thermal power plant :*

Assume R for air as 287 J/kg C.
Steam condensed = 40,000 kg/hr
Temperature of steam in condenser = 40° C
Dryness of steam entering condenser = 0.85
Air leakage in the condenser = 140 kg/hr
Temperature of condenser = = 35° C
Temperature at the suction of air pump = 32° C
Barometer reading = 76 cm of Hg
Maximum rise in cooling water temperature = 12° C
Find : (i) Vacuum gauge reading in condenser in cm of Hg.
(ii) capacity of dry-air pump in m^3/hr.
(iii) Quantity of cooling water passed through condenser in tons/hr.

[G.U., Dec. '04, 8 marks]

Solution : Given data

R = 287 J/kg°C, m_s = 40,000 kg/h, T_s = 40°C, x = 0.85, m_a = 140 kg/h, T_h = 35°C,

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Temperature of air = 32°C, Barometer reading = 76 cm of Hg, $(T_o - T_i) = 12°C$

Vacuum gauge reading = ?, Air pump capacity = ?, m_w = ?

(i) Vacuum gauge reading in condenser,

Total pressure in the condenser is given by

$$p_c = p_s + p_a$$

where p_s at 40 °C saturation temperature, from steam tables

$p_s = 0.0738$ bar, and $v_g = 19.52\ m^3/kg$

Volume of 40,000 kg of steam is given by

$$V = 40,000 \times xv_g = 40,000 \times 0.85 \times 19.52$$
$$= 663680\ m^3/h$$

The same volume is also occupied by 140 kg of air, therefore, the pressure of air in the condenser is given by

$$p_a = \frac{m_a RT}{V} = \frac{140 \times 287 \times (273+40)}{663680} = 18.95\ N/m^2$$
$$= 18.95 \times 10^{-5}\ \text{bar}$$

$$\therefore p_c = 0.0738 + 18.95 \times 10^{-5} = 0.07399\ \text{bar}$$

∴ Vacuum gauge reading in condenser = Std. Atmospheric pressure $- p_c$

$$= 1.0132 - 0.07399 = 0.9392\ \text{bar}$$

$$= \frac{0.9392}{1.333 \times 10^{-2}} = \mathbf{70.458\ cm\ of\ Hg.}\ \textit{Ans}$$

(ii) Capacity of dry-air pump.

From steam tables, partial pressure of steam at air-pump suction is the saturation pressure of steam corresponding to 32°C,

$p_{s_1} = 0.0476$ bar

∴ Partial pressure of air, $p_{a_1} = p_c - p_{s_1} = 0.07399 - 0.0476$

$$= 0.02639\ \text{bar}$$

Volume of air at 32°C and 0.02639 bar is given by

$$V_1 = \frac{m_a RT}{p_{a_1}} = \frac{140 \times 287 \times (273+32)}{0.02639 \times 10^5} = 4643.766\ m^3/h$$

∴ Air pump capacity = **4643.766 m³/h** *Ans*

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(iii) Quantity of cooling water passed through condenser,

From steam tables, at 40°C saturation temperature

$$h_f = 167.6 \text{ kJ/kg}, \; h_{fg} = 2406.7 \text{ kJ/kg}$$

We know that total heat entering steam,

$$h = h_f + x h_{fg} = 167.6 + 0.85 \times 2406.7 = 2213.295 \text{ kJ/kg}$$

Enthalpy of condensate corresponding to temperature of condensate = 35°C,

$$h_{f_1} = 146.7 \text{ kJ/kg (from steam table)}$$

We also know that mass of cooling water,

$$m_w = \frac{m_s (h - h_{f_1})}{c_w (T_o - T_i)} = \frac{40{,}000(2213.295 - 146.7)}{4.186 \times 12}$$

$$= 1645640.229 \text{ kg/h} = \frac{1645640.229}{1000} \cong \mathbf{1645.64 \text{ tons/h}} \quad \textit{Ans}$$

Problem 7.7 *A surface condenser fitted with separate air and water extraction pumps, has a portion of the tubes near the air pump suction screened off from the steam so that the air is cooled below the condensate temperature. The steam enters the condenser at 35°C and the condensate is removed at 34°C. The air removed has a temperature of 33°C. If the total air infiltration from all sources together is 5 kg/h, calculate the volume of air handled by the air pump per hour. What would be the corresponding value of the air handled if a combined air and condensate pump was employed ? Assume uniform pressure in the condenser.*

Solution : Given data

$$T_s = 35°\text{C}, \; T_c = 34°\text{C}, \; T_a = 33°\text{C}, \; m_a = 5 \text{ kg/h}, \; V_a = ?, \; V_a' = ?$$

(i) Volume of air handled by the air pump per hour

From steam tables, at 35°C, $p_c = 0.0563$ bar

and pressure of steam at the air pump suction, corresponding to 33°C, $p_s = 0.0503$ bar

As per Dalton's law, pressure of air at the air pump suction

$$p_a = p_c - p_s = 0.0563 - 0.0503 = 0.006 \text{ bar}$$

$$= 0.006 \times 10^5 = 600 \text{ N/m}^2$$

We know that volume of air handled by the air pump,

$$V_a = \frac{m_a R T_a}{p_a} = \frac{5 \times 287 \times (273 + 33)}{600} = \mathbf{731.85 \text{ m}^3\text{/h}} \quad \textit{Ans}$$

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(ii) Volume of air handled when a combined air and condensate pump is employed.

From steam tables, corresponding to a condensate temperature of 34°C, we find that pressure of steam

$$p_s = 0.0532 \text{ bar}$$

$\therefore$ Pressure of air, $p_a = p_c - p_s = 0.0563 - 0.0532 = 0.0031$ bar

$$= 0.0031 \times 10^5 = 310 \text{ N/m}^2$$

We know that volume of air handled,

$$V_a' = \frac{m_a R T_c}{p_a} = \frac{5 \times 287 \times (273+34)}{310}$$

$$= \mathbf{1421.113 \text{ m}^3/\text{h}}$$ **Ans**

Problem 7.8 *During trial of steam condenser, following data were observed.*

Condenser vacuum recorded = 70 cm of Hg, Barometric reading = 76.5 cm of Hg, Mean condenser temperature = 36°C, Hot well temperature = 29°C, condensate formed per hour = 1750 kg; circulating cooling water inlet temperature = 15°C, circulating cooling water outlet temperature = 27°C, cooling water circulated per hour = 76 tons; Air leakage = 1 kg/ton of steam. Determine : (i) vacuum efficiency, (ii) condenser efficiency, (iii) Quality of steam entering the condenser, (iv) capacity of wet extraction pump in m^3/h.

Solution : Given data

Condenser vacuum = 70 cm of Hg, Barometer reading = 76.5 cm, $T_c = 36°C$, $T_h = 29°C$, $m_s = 1750$ kg/h, $T_i = 15°C$, $T_o = 27°C$, $m_w = 76$ tons/h = 76000 kg/h, $m_a = 1$ kg/ton of steam, $\eta_v = ?$, $\eta_c = ?$, $x = ?$, capacity of pump = ?

(i) Vacuum efficiency

Absolute pressure in condenser,

p_c = Barometer reading – condenser vacuum

$= 76.5 - 70 = 6.5$ cm of Hg $= 6.5 \times 1.333 \times 10^{-2}$ bar

$= 0.08664$ bar

From steam tables, partial pressure of steam at $T_c = 36°C$

$$p_s = 0.0595 \text{ bar} = \frac{0.0595}{1.333 \times 10^{-2}} = 4.4636 \text{ cm of Hg.}$$

$\therefore$ Partial pressure of air, $p_a = p_c - p_s$

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$= 0.08664 - 0.0595$

$= 0.02714$ bar

Ideal vacuum = Standard barometer reading – Ideal vacuum

$= 76 - p_s = 76 - 4.4636 = 71.5364$ cm of Hg.

Actual vacuum = Standard barometer reading – (Barometer reading – condenser vacuum)

$= 76 - (76.5 - 70) = 69.5$ cm of Hg.

$\therefore$ Vacuum efficiency, $\eta_v = \dfrac{\text{Actual vacuum}}{\text{Ideal vacuum}} = \dfrac{69.5}{71.5364}$

$= 0.9715 = \mathbf{97.15\ \%}$ ***Ans***

(ii) Condenser efficiency

From steam tables, corresponding to condenser pressure of $p_c = 0.08664$ bar,

$T_s = 42.7^{\circ}C$

$\therefore$ Condenser efficiency, $\eta_c = \dfrac{T_o - T_i}{T_s - T_i} = \dfrac{27 - 15}{42.7 - 15}$

$= 0.4332 = \mathbf{43.32\ \%}$ ***Ans***

(iii) Quality of steam entering to the condenser, x;

From steam tables at $T_c = 36^{\circ}C$, we have

$h_f = 150.9$ kJ/kg, $h_{fg} = 2416.2$ kJ/kg,

$V_g = 23.94\ m^3/kg$

and at hotwell temperature $T_h = 29^{\circ}C$, we have

$h_{f_1} = 121.6$ kJ/kg.

We know that total heat entering steam

$h = h_f + xh_{fg} = 150.9 + x \times 2416.2$

We also know that mass of cooling water (m_w)

$$m_w = \frac{m_s(h - h_{f_1})}{c_w(T_o - T_i)};\quad 76000 = \frac{1750(150.9 + x \times 2416.2 - 121.6)}{4{,}186 \times (27 - 15)}$$

$\therefore\ x = \mathbf{0.8907}$ ***Ans***

(iv) Capacity of wet extraction pump, V;

Air leakage, $m_a = 1 \times \dfrac{1750}{1000} = 1.75$ kg/h

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Total mass to be extracted by pump, $m = m_a + m_s = 1.75 + 1750$

$= 1751.75$ kg/hr

$\therefore V = m \times xv_g = 1751.75 \times 0.8907 \times 23.94$

$= \mathbf{37353.192\ m^3/h}$ **Ans**

Problem 7.9 *A jet condenser is required to condense 4500 kg/h of steam. 330 m^3 of injection water is used per hour. Initial temperature of cooling water is 25°C. The volume of air at atmospheric pressure dissolved in injection water is 5% of the volume of water. The amount of air entering the condenser with steam is 1 kg per 3300 kg of steam. The vacuum in the air pump suction is 680 mm of Hg. When the barometer reads 760 mm of Hg and the temperature of the condensate is 35°C, determine the suction capacity of the air pump in m^3/min to remove air and water from the condenser. Assume volumetric efficiency of pump as 85%.*

Solution : Given data

$m_s = 4500\ kg/h$, volume of injection water $V_w = 330\ m^3/h$, $T_i = 25°C$,

Volume of air in injection water = 5% of 330 m^3/h

mass of air entering with steam = 1 kg/3300 kg of steam.

Condenser vacuum = 680 mm of Hg, Barometer reading = 760 mm of Hg

Condensate temperature, $T_c = 35°C$, $\eta_{vol} = 85\%$, Pump capacity = ?

Absolute pressure in condenser,

p_c = Barometer reading – condenser vacuum

$= 760 - 680 = 80$ mm of Hg $= 80 \times 1.333 \times 10^{-3}$ bar

$= 0.1066$ bar

From steam tables, corresponding to condensate temperature 35°C,

$p_s = 0.0563$ bar and $v_g = 25.22\ m^3/kg$.

$\therefore$ Partial pressure of air, $p_a = p_c - p_s = 0.1066 - 0.0563$

$= 0.0503$ bar

Mass of air associated with steam, $m_a = \frac{1}{3300} \times 4500 = 1.3636\ kg/h$

Volume of air associated with steam to be condensed,

$$V_{a_1} = \frac{m_a R T_a}{p_a} = \frac{1.3636 \times 287 \times (273+35)}{0.0503 \times 10^5}$$

$= 23.963\ m^3/h$

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Total volume of air to be handled,

V_a = Volume of air in (condensed steam + injection water)

$= 23.963 + 0.05 \times 330 = 40.463 \text{ m}^3/h$

Remaining mass of steam to be condensed,

m_s' = Mass of steam – mass of air associated

$= 4500 - 1.3636 = 4498.636 \text{ kg/h}$

Specific volume of water at 35°C from steam tables,

$v_f = 0.001006 \text{ m}^3/\text{kg}$

$\therefore V_f = m_s' \cdot v_f = 4498.636 \times 0.001006 = 4.526 m^3/h$

Total volume of water in condenser,

$V_w' = V_w + V_f = 330 + 4.526 = 334.526 \text{ m}^3/\text{h}$

Volume of water and air to be handled by suction pump,

$V = V_w' + V_a = 334.526 + 40.463 = 374.989 \text{ m}^3/\text{h}$

∴ Handling capacity of suction pump in m^3/min

$$\therefore V_{cs} = \frac{V}{\eta_{vol}} = \frac{374.989}{60} \times \frac{1}{0.85}$$

$$= \mathbf{7.3527 \text{ m}^3/\text{min}}$$ ***Ans***

Problem 7.10 *Dry steam is condensed at a rate of 7000 kg/h and the air leakage amounts to 14 kg/h. The air pump suction is screened off. The exhaust steam temperature is 32°C. The condensate temperature is 30°C and the temperature at the air pump suction is 25°C. Determine (i) the volume of air handled by the pump, (ii) the mass of steam condensed in the air cooler section per hour, (iii) mass of water vapour carried away by air, and (iv) the percentage reduction in the air pump capacity due to cooling of air.*

Solution : Given data (refer Fig. 7.13)

$m_s = 7000$ kg/h, $m_a = 14$ kg/h, $T_c = 32°C$, condensate temperature = 30°C, Air pump suction temperature = 25°C.

(i) The volume of air handled by the pump

From steam tables, corresponding to 32°C temperature

$p_s = 0.0476$ bar, $v_g = 29.54 \text{ m}^3/\text{kg}$

∴ Volume of steam at inlet per hour, $V = m_s \times v_g$ (∵ dry and sat. steam)

$= 7000 \times 29.54 = 206780 \text{ m}^3/\text{h}$

Air occupies same volume as steam

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$\therefore$ Volume of air at inlet $V_a = 206780\ m^3/h$

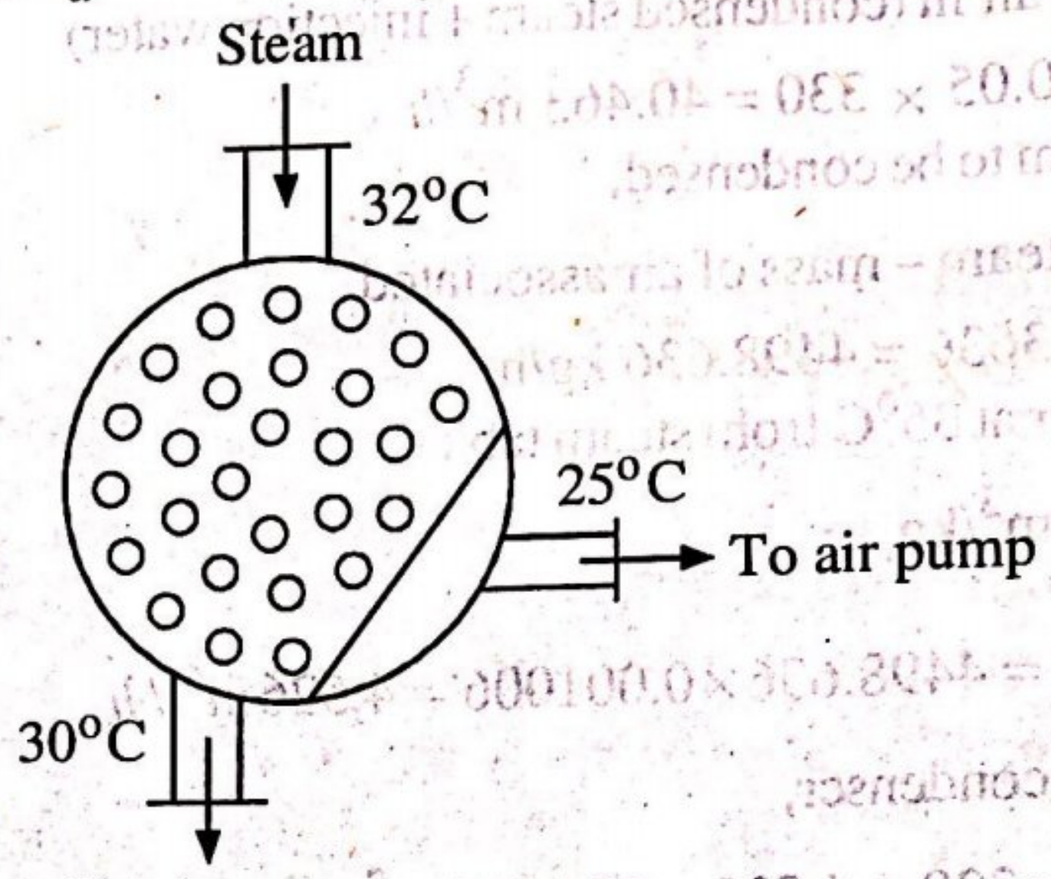


Fig. 7.13

$\therefore$ Partial pressure of air at inlet, $p_a = \dfrac{m_a R T_c}{V_a}$

$$= \frac{14 \times 287 \times (32+273)}{206780} = 5.93\ N/m^2$$

$$= 5.93 \times 10^{-5}\ \text{bar}$$

$\therefore$ Total pressure in condenser, $p_c = p_s + p_a = 0.0476 + 5.93 \times 10^{-5}$

$= 0.04766$ bar

From steam tables, corresponding to air pump suction temperature = 25°C.

Partial pressure of steam $p_s = 0.0317$ bar, $v_g = 43.36\ m^3/kg$.

$\therefore$ Partial pressure of air at air pump suction, $p_a = p_c - p_s$

$\therefore p_a = 0.04766 - 0.0317 = 0.01596$ bar

$\therefore$ Volume of air per hour at air pump inlet

$$V_a = \frac{m_a R T_a}{p_a} = \frac{14 \times 287 \times (25+273)}{0.01596 \times 10^5} = \mathbf{750.228\ m^3/h.}$$ **Ans**

(ii) The mass of steam condensed at cooler section

Steam occupies same volume as air.

$\therefore$ Volume of steam at cooler section = 750.228 m^3/h

$\therefore$ Mass of steam at air pump inlet,

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$$= \frac{\text{Volume of steam}}{v_g} = \frac{750.228}{43.36} = 17.302\, kg/h$$

From steam tables, corresponding to condensate temperature of 30°C

$$p_s = 0.0425 \text{ bar}, \; v_g = 32.89 \text{ m}^3/\text{kg}$$

$\therefore$ Partial pressure of air, $p_a = p_c - p_s = 0.04766 - 0.0425$

$$= 5.16 \times 10^{-3} \text{ bar}$$

$\therefore$ Volume of air, $V_a = \frac{m_a RT}{p_a} = \frac{14 \times 287 \times (30 + 273)}{5.16 \times 10^{-3} \times 10^5}$

$$= 2359.407 \text{ m}^3/\text{hr}$$

$\therefore$ Mass of steam condensed in cooler section = 71.736 – 17.302

$$= \mathbf{54.434 \text{ kg/h}} \quad \textbf{Ans}$$

(iii) Mass of water vapour carried away by air, m_w;

$$m_w = \frac{V_a \text{ at air pump suction}}{v_g \text{ at air pump suction}} = \frac{750.228}{43.36} = \mathbf{17.302 \text{ kg/h}} \quad \textbf{Ans}$$

(iii) Percentage reduction in the air pump capacity due to cooling of air

$$= \frac{2359.407 - 750.228}{2359.407} \times 100 = \mathbf{68.20\%} \quad \textbf{Ans}$$

Problem 7.11 *The following observations were recorded during a test on a steam condenser :*

Barometer reading = 764 mm of Hg
Condenser vacuum = 712 mm of Hg
Mean condenser temperature = 36°C
Condensate temperature = 29°C
Condensate collected per hour = 2 tonnes
Quantity of cooling water per hour = 60 tonnes
Temperature of cooling water at inlet = 12°C
Temperature of cooling water at outlet = 27°C

Determine : (i) vacuum corrected to the standard barometer reading, (ii) vacuum efficiency of the condenser, (iii) undercooling of the condensate, (iv) condenser efficiency, (v) quality of the steam entering the condenser, (vi) mass of air per m^3 of condenser volume, and (vii) mass of air per kg of uncondensed steam.

Solution : Given data :

Barometer reading = 764 mm of Hg, Condenser vacuum = 712 mm of Hg,

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$T = 36^{\circ}C = 309$ K, $T_c = 29^{\circ}C$, $m_s = 2$ t/h $= 2000$ kg/h, $m_w = 60$ t/h $= 60{,}000$ kg/h,

$T_i = 12^{\circ}C$, $T_o = 27^{\circ}C$

(i) Vacuum corrected to the standard barometer reading,

Absolute pressure in the condenser

= Barometer reading – condenser vacuum

= 764 – 712 = 52 mm of Hg

The vacuum corrected to the standard barometer reading

= 760 – 52 = **708 mm of Hg.** ***Ans***

(ii) Vacuum efficiency of the condenser

From steam tables, corresponding to the mean condenser temperature of 36°C,

$$p_s = 0.0595 \text{ bar} = \frac{0.0595}{0.001333} = 44.636 \text{ mm of Hg.}$$

We know that ideal vacuum

= Barometer pressure – Ideal pressure

= 764 – 44.636 = 719.364 mm of Hg

and vacuum efficiency, $\eta_v = \dfrac{\text{Actual vacuum}}{\text{Ideal vacuum}} = \dfrac{712}{719.364}$

$= 0.9897 = \mathbf{98.97\%}$ ***Ans***

(iii) Undercooling of the condensate

We know that undercooling of the condensate

= Mean condenser temperature – condensate temperature

$= 36 - 29 = \mathbf{7^{\circ}C}$ ***Ans***

(iv) Condenser efficiency

We have already found that pressure in the condenser

$p_c = 764 - 712 = 52$ mm of Hg $= 52 \times 0.001333 = 0.0693$ bar

From steam tables, corresponding to a pressure of 0.0693 bar,

$T_s = 39^{\circ}C$

$\therefore$ Condenser efficiency, $\eta_c = \dfrac{T_o - T_i}{T_s - T_i} = \dfrac{27 - 12}{39 - 12} = 0.5555 = \mathbf{55.55\%}$ ***Ans***

(v) Quality of steam entering the condenser

Let x = Quality of steam entering the condenser.

From steam tables, corresponding to a pressure of 0.0693 bar, we find that

$h_f = 163.4$ kJ/kg, and $h_{fg} = 2409.1$ kJ/kg

and corresponding to condensate temperature of 29°C,

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heat in condensate, $h_{f_1} = 121.6$ kJ/kg

We know that total heat of entering steam

$$h = h_f + xh_{fg} = 163.4 + x \times 2409.1$$

We also know that mass of cooling water (m_w)

$$m_w = \frac{m_s(h - h_{f_1})}{c_w(T_o - T_i)}$$

$$\therefore\ 60000 = \frac{2000 \times (163.4 + x \times 2409.1 - 1216)}{4.2(27 - 12)}$$

$$= 31.746 \times (41.8 + x \times 2409.1)$$

$$\therefore\ x = \mathbf{0.767}$$ **Ans**

(vi) Mass of air per m^3 of condenser volume

As per Dalton's law, absolute pressure of air, $p_a = p_c - p_s$

$$= 0.0693 - 0.0595 = 9.8 \times 10^{-3} \text{ bar} = 980 \text{ N/m}^2$$

$\therefore$ Mass of air per m^3 of condenser volume,

$$m_a = \frac{p_a v}{RT} = \frac{980 \times 1}{287 \times 309} = \mathbf{0.011\ kg}$$ **Ans**

(vii) Mass of air per kg of uncondensed steam

From steam tables, corresponding to mean condenser temperature of 36°C,

specific volume of steam, $v_g = 23.94\ m^3/kg$

Thus, air associated with 1 kg of steam at 36°C will occupy the same volume i.e. 23.94 m^3/kg.

$\therefore$ Mass of air per kg of uncondensed steam

$$m_a = \frac{p_a v_g}{RT} = \frac{980 \times 23.94}{287 \times 309} = \mathbf{0.2645\ kg}$$ **Ans**

Problem 7.12 *A steam turbine of 20,000 kW capacity requires 5 kg. of steam per hour pe kW. The quantity of air leakage into the condenser is 1 kg per 1000 kg of steam used by turbine. The vacuum in the condenser is 70 cm of Hg when the barometer reads 76 cm of Hg. The temperature at the suction of the air pump is 30° C. The surface condenser used is fitted with separate condensate pump and air pump. Find :*

(a) The capacity of the air pump requires per minute.

(b) The mass of water vapor carried by air in kg/hr.

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(c) The quantity of cooling water required per minute in tons if the rise in temperature of water is limited to 8^oC.

The quality of steam entering the condenser is 0.9 dry and there is no under cooling in the condenser.

Properties of steam at 30^oC.

Absolute pressure = 0.04325 bar

Enthalpy of liquid h_f = 125.8 kJ/kg

Enthalpy of evaporation h_{fg} = 2438.1 kJ/kg

Specific volume = 32.8 m^3/kg. **[Dec. '11, 7 marks]**

Solution : Given data

Steam turbine capacity = 20,000 kW, m_s = 5 kg/h per kW, m_a = 1 kg per 1000 kg of steam, p_c = 70 cm of Hg, p_b = 76 cm of H_g, $T_a = 30^oC$, $T_o - T_i = 8^oC$, $x = 0.9$, capacity of air pump = ?, m_v = ?, m_w = ?.

(a) The capacity of the air pump :

Absolute pressure inside the condenser p_c

= Barometer reading – condenser vacuum

$= 76 - 70 = 6$ cm of $H_g = 6\times10^{-2}\times13.6\times1000\times9.81$

$= 8004.96\ N/m^2 \cong 0.08$ bar

From steam tables at 30°C, $p_s = 0.04325$ bar

$\therefore$ Partial pressure of air, $p_a = p_c - p_s = 0.08 - 0.4325 = 0.03675$ bar

Air leakage into the condenser per minute,

$$m_a = 20000\times5\times\frac{1}{1000}\times\frac{1}{60} = 1.67\ kg/\min$$

Using characteristic gas equation,

$$p_aV = m_aRT$$

$$\therefore 0.03675\times10^5\times V = 1.67\times287\times(30+273)$$

$$\therefore V = 39.517\ m^3/\min$$

$\therefore$ Capacity of air pump = **39.517 m^3/min** Ans

(b) The mass of water vapor carried by air :

Specific volume of steam at 30°C, $v_g = 32.8\ m^3/kg$

Mass of water vapour carried by air, $m_v = \frac{V}{v_g} = \frac{39.517\times60}{32.8}$

= 72.287 kg/h Ans

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(c) The quantity of cooling water required :

$$m_w = \frac{m_s\left(h_f + xh_{fg} - h_{f1}\right)}{C_w\left(T_o - T_i\right)}$$

If there is no under cooling then, $h_{f1} = h_f$

and at 30°C saturated temperature, $h_{fg} = 2438.1\ kJ/kg$

Mass of steam is given by $m_s = \frac{20000 \times 5}{60}\ kg/\min$

$$\therefore m_w = \frac{m_s \times xh_{fg}}{C_w\left(T_o - T_i\right)} = \frac{20000 \times 5}{60} \times \frac{0.9 \times 2438.1}{4.186 \times 8}$$

$= 109207.776$ kg/min $=$ **109.2077 tons/min.** ***Ans***

7.12 Necessity of cooling ponds and cooling towers [Dec. '11]

The cooling water is required to be circulated in the condenser inorder to absorb the heat from steam. As the cooling water takes the latent heat of steam in the condenser, the temperature of the water increases. The hot circulating water coming out of the condenser cannot be used again in a closed system without precooling. This is because, the hot water coming out if used again will not be able to absorb the heat as temperature of cooling water reaches near to saturation temperature of steam at condenser pressure and the condenser vacuum cannot be maintained. Therefore it is absolutely necessary to precool the water coming out of condenser before using again.

7.13 Condenser water cooling systems

The condenser cooling water required in an open system is about 50 times the mass flow of steam while in closed loop using cooling towers, it is about 5 to 7.5 kg/kwh. Therefore in large steam power plants the cooling water required, say for a 1000 MW plant may be of the order of 100 thousand tons of water per day. The cooling systems which are commonly used in practice according to the availability of the water are listed below :

(1) Open or once through system.

(2) Closed cooling system.

(3) Mixed cooling system.

(1) Open or once through system : [June '11]

This system is used when there is a large source of water available. Water is taken from a natural body of water like a river, lake, or ocean and pumped through the condenser, where it is heated, and then discharged back to the downward side of the source. The

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temperature of the discharged water should be kept within safe limits to prevent harm to fishes.

The arrangement of the open system is shown in Fig. 7.14. The position of inlet and outlet should be chosen in such a way that there should not be recirculation of hot water which impairs the efficiency of the condensing plant. This type of cooling system can be used only where required quantity of water is available throughout the year.

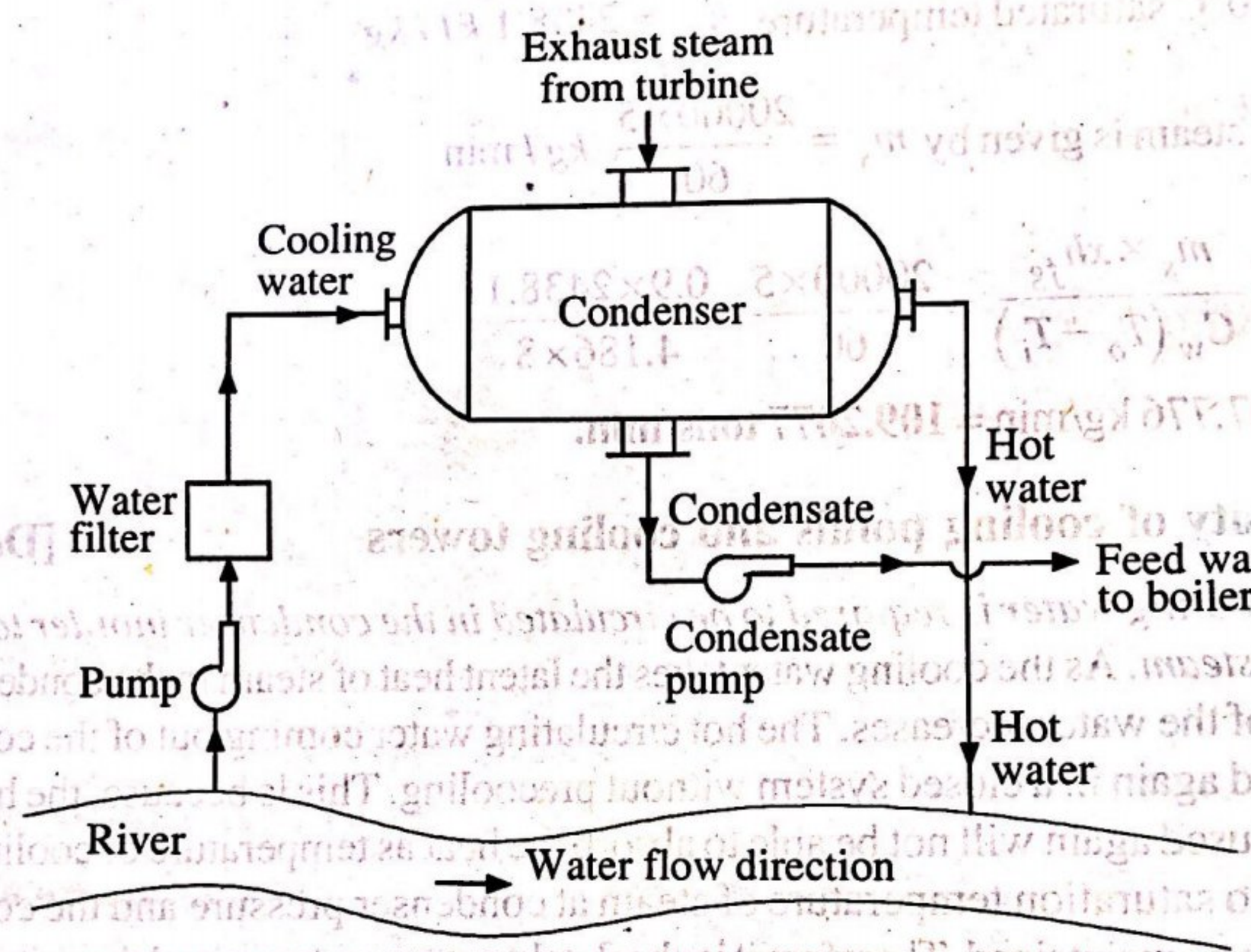


Fig. 7.14 Open cooling system

(2) Closed cooling system :

In closed loop cooling system, warm water from the condenser is passed through a cooling device like a cooling tower or spray pond and the cooled water is then pumped back for condenser circulation. With this system of cooling also, an external source of water is needed to replace tower evaporation and carry-over losses. The quantity of water lost by evaporation and carry-over varies between 2 to 5% of that circulated depending upon the design of tower

The arrangement of this cooling system is shown in Fig. 7.15. Such system is usually employed when the sufficient cooling water required for condensing the steam from lake or river is not available.

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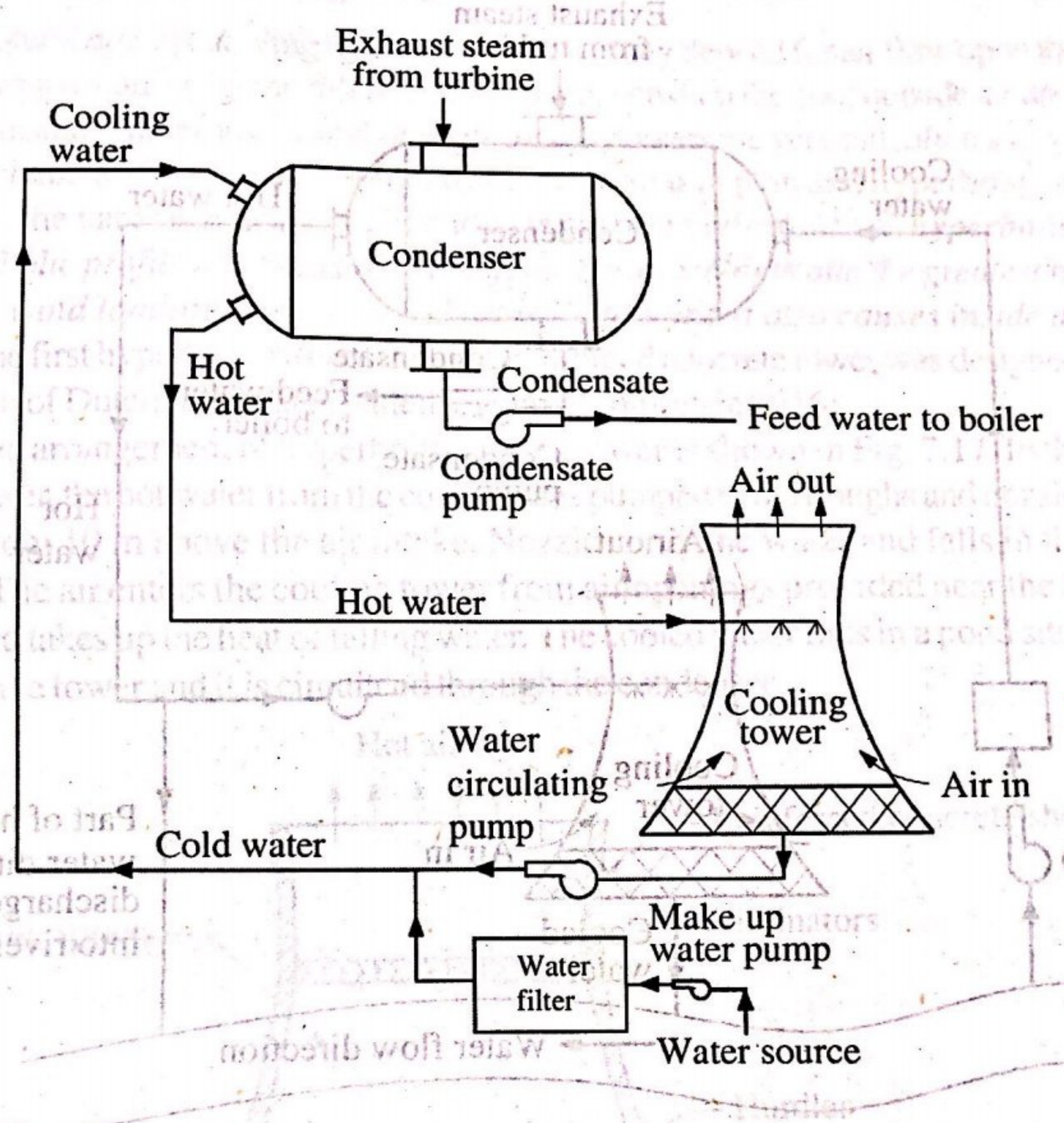


Fig. 7.15 Closed cooling system

(3) Mixed cooling system :

The combination of both open and closed cooling system is called the mixed cooling system. The arrangement of this system is shown in Fig. 7.16. In this system, using the river and cooling towers overcomes the difficulty of re-circulation and meet the requirements of a fishery board on a fairly small river. The hot water received from condenser is divided in two parts, one part of it is directly discharged into the river as in case of open system and the remainder part is passed into the cooling tower for its cooling and then discharged into the river. The cooling water is supplied from river directly into the condenser.

- **Advantages :**
  - (1) The quantity of cooling water required is reduced as re-circulation is eliminated.
  - (2) The size of cooling towers can be reduced.
  - (3) It increases turbine plant efficiency.

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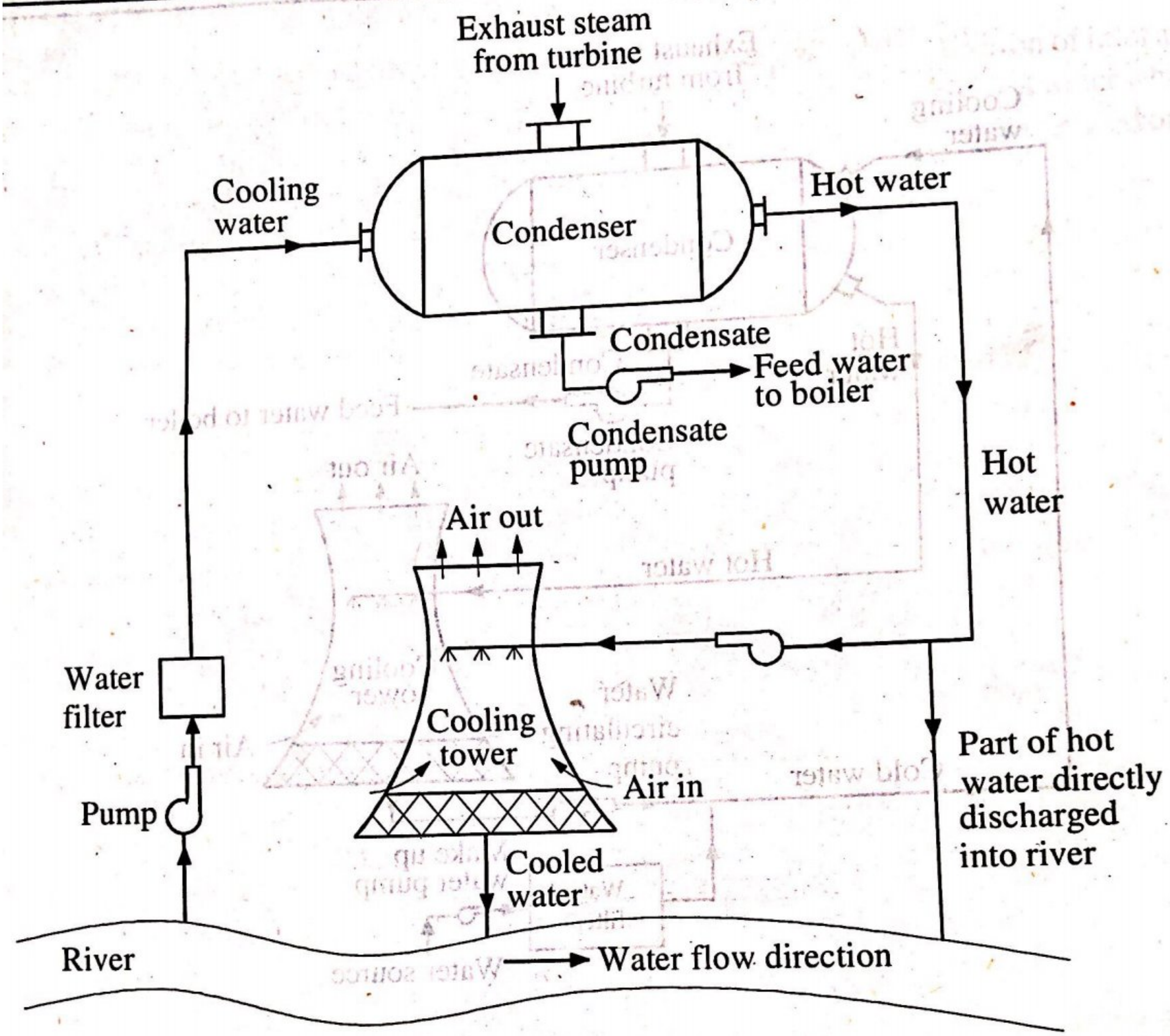


Fig. 7.16 Mixed cooling system

7.14 Types of cooling towers

Cooling towers cool the warm water discharged from the condenser and feed the cooled water back to the condenser. They, thus, reduce the cooling water demand in the power plant. The present trend is to locate the power plant near the load centre with the use of large and highly efficient cooling towers.

The different types of cooling towers are as follows :

(1) Natural draught-cooling towers

(2) Mechanical cooling towers

(a) Forced draught cooling towers

(b) Induced draught cooling towers

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7.14.1 Natural draught cooling towers [Dec. '11]

Natural draught cooling towers use no fans. They depend for air flow upon the natural driving pressure caused by the difference in density between the cool outside air and the hot, humid air inside. Therefore, natural draught cooling towers are very tall, often a few hundred feet. The shape of the body of cooling tower is circular in plan and hyperbolic in profile. Therefore, the natural draught cooling towers are often referred to as ***hyperbolic towers***. *The hyperbolic profile has been found to offer superior strength and the greatest resistance to outside wind loading compared with other forms and it also causes inside air flow.*

The first hyperbolic natural draught reinforced concrete tower was designed by Prof. Van Itesson of Dutch state and installed at Emma Collisey in 1916.

The arrangement of hyperbolic cooling tower is shown in Fig. 7.17. In this type of cooling tower, the hot water from the condenser is pumped to the troughs and nozzles situated at the bottom 10 m above the air intake. Nozzle spray the water and falls in the form of droplets. The air enters the cooling tower from air openings provided near the base, rises upward and takes up the heat of falling water. The cooled water falls in a pond situated at the bottom of the tower and it is circulated through the condenser.

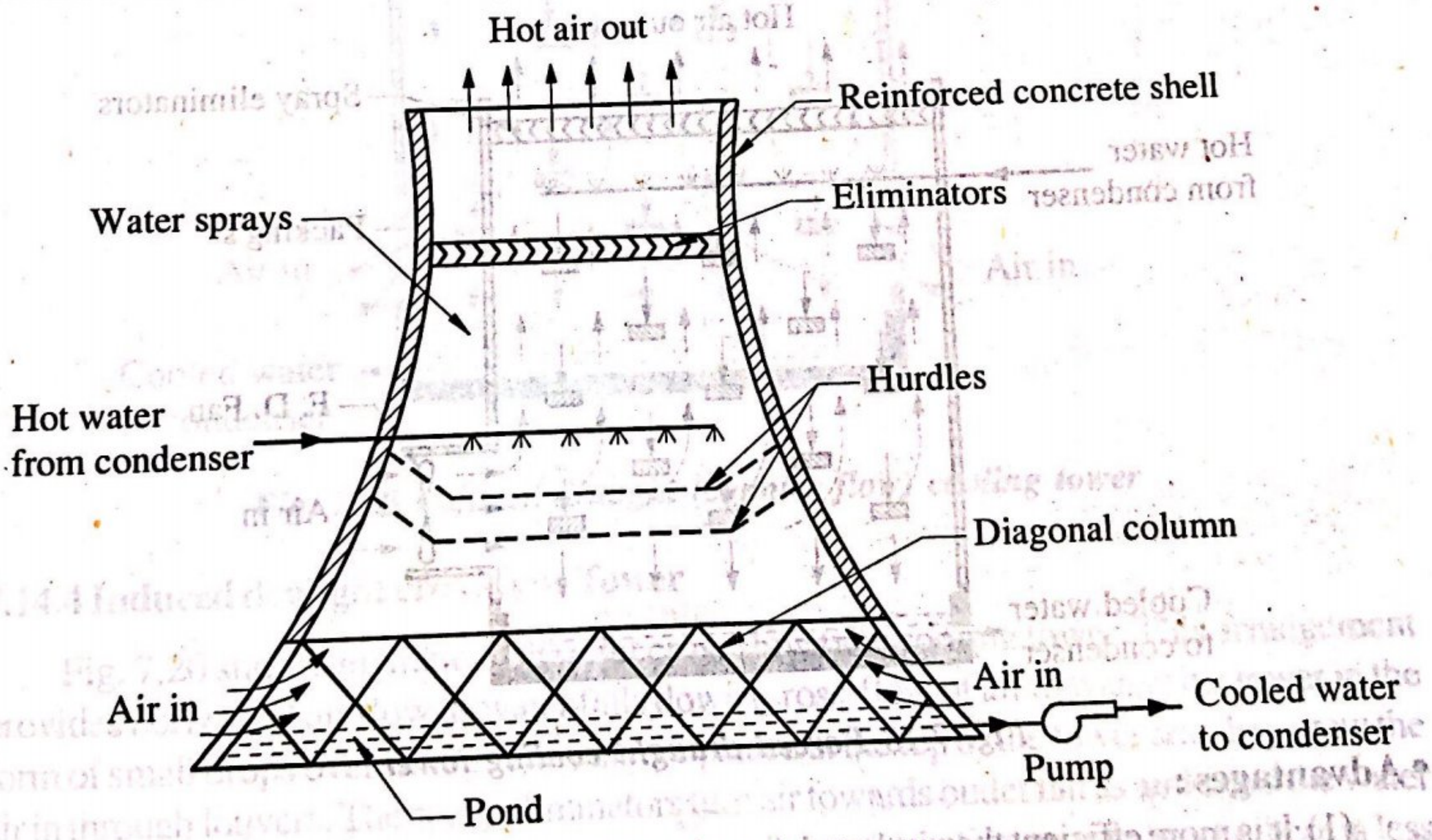


Fig. 7.17 Hyperbolic cooling tower

- **Advantages :**
  (1) Its operating and maintenance costs are less since no fans are needed.
  (2) Hyperbolic tower's chimney shape creates its own draft assuring efficient operation even when there is no wind.

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(3) They require considerable less ground area.
(4) The enlarged top of the tower allows water to fall out of suspension.
(5) It gives more or less trouble free operation.
(6) Longer in life.

- **Disadvantages :**
  (1) Its capital cost is considerably high since the towers may be as high as 125 m and 100m in diameter at the base.
  (2) Its performance varies with the seasonal changes in Dry Bulb Temperature (DBT) and Relative Humidity (RH) of air.

7.14.2 Forced draught cooling tower

The forced draught cooling tower is shown in Fig. 7.18. In this tower, forced draught fan is located at the bottom of the cooling tower. The hot water from condenser is supplied at the top of the cooling tower which is sprayed through nozzles. The air is blown by the fan up through the descending water. The rising air cools the incoming water. The spray eliminators are used to remove the entrained water in air.

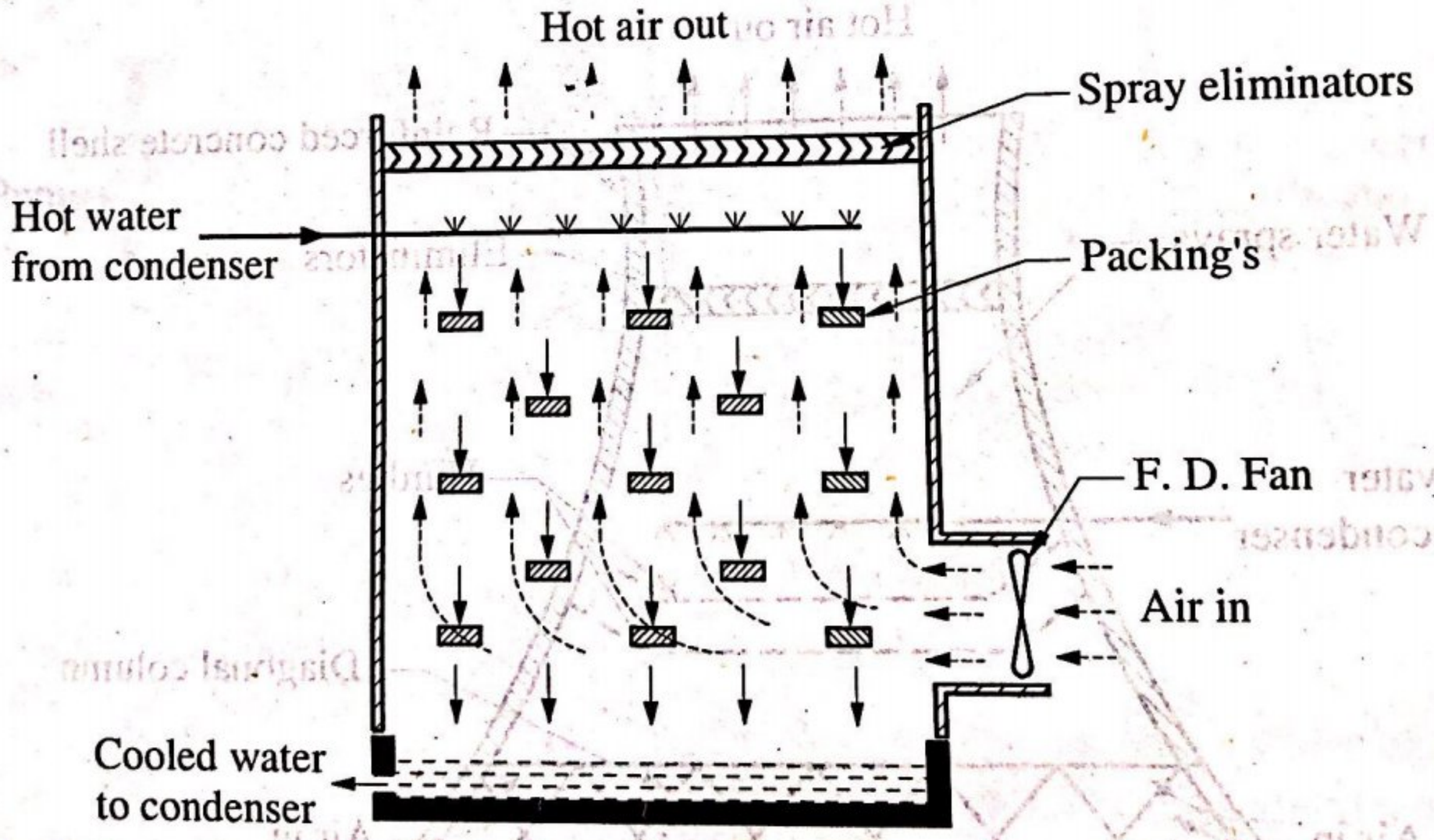


Fig. 7.18 Forced draught cooling tower

- **Advantages :**
  (1) It is more efficient than induced draught cooling tower.
  (2) It's fan handles dry air, problems of blade erosion are avoided.
  (3) It is more safe as fan is located on the ground level.
  (4) Vibration and noise are less since mechanical equipments are set on a solid foundation.
- **Disadvantages :**
  (1) There is possibility of recirculation of hot humid exhaust air coming out from the top of the tower.

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(2) The fan size is limited to 4 metres.

(3) There is possibility of ice formation on fan blades in cold weather, thus the fan blades may break.

7.14.3 Induced draught counter flow tower

Fig. 7.19 shows an induced draught cooling tower. In these towers, the fans are placed at the top of the tower and they draw the air in through louvers extending all around the tower at its base.

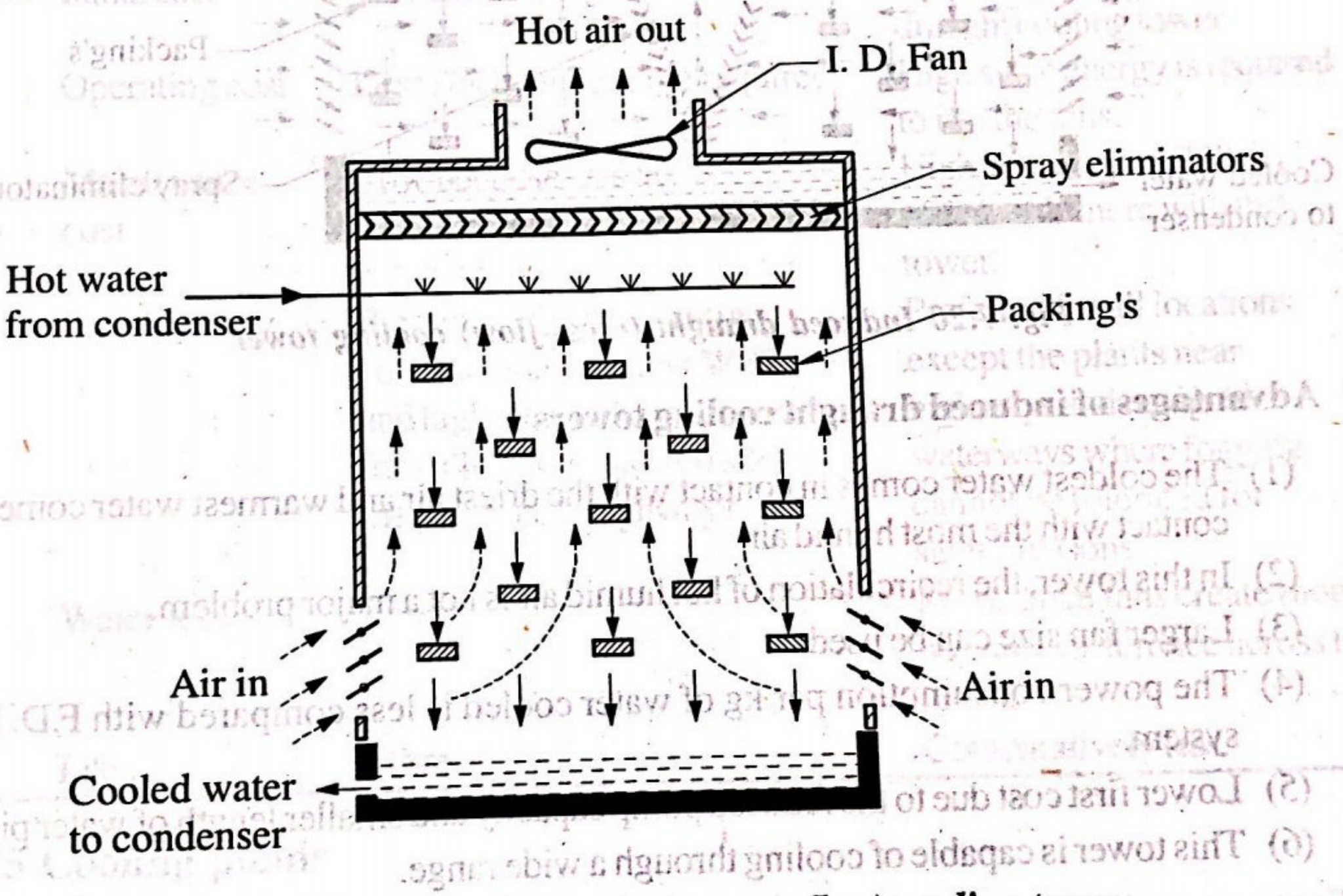


Fig. 7.19 Induced draught (counter flow) cooling tower

7.14.4 Induced draught cross-flow Tower

Fig. 7.20 shows an induced draught cross flow type cooling tower. This arrangement provides horizontal air flow as water falls down (cross-flow of air to water) the tower in the form of small drops over filling. The fans are placed at the top of the tower and they draw the air in through louvers. The spray eliminators turn air towards outlet fan as air leaves the water sprays. The special feature of this arrangement is lower air static pressure loss as there is less resistance to air flow and resulting in lower fan power requirement.

The counterflow arrangement is more efficient than cross-flow from a thermodynamic point of view, because its enthalpy potential difference is higher.

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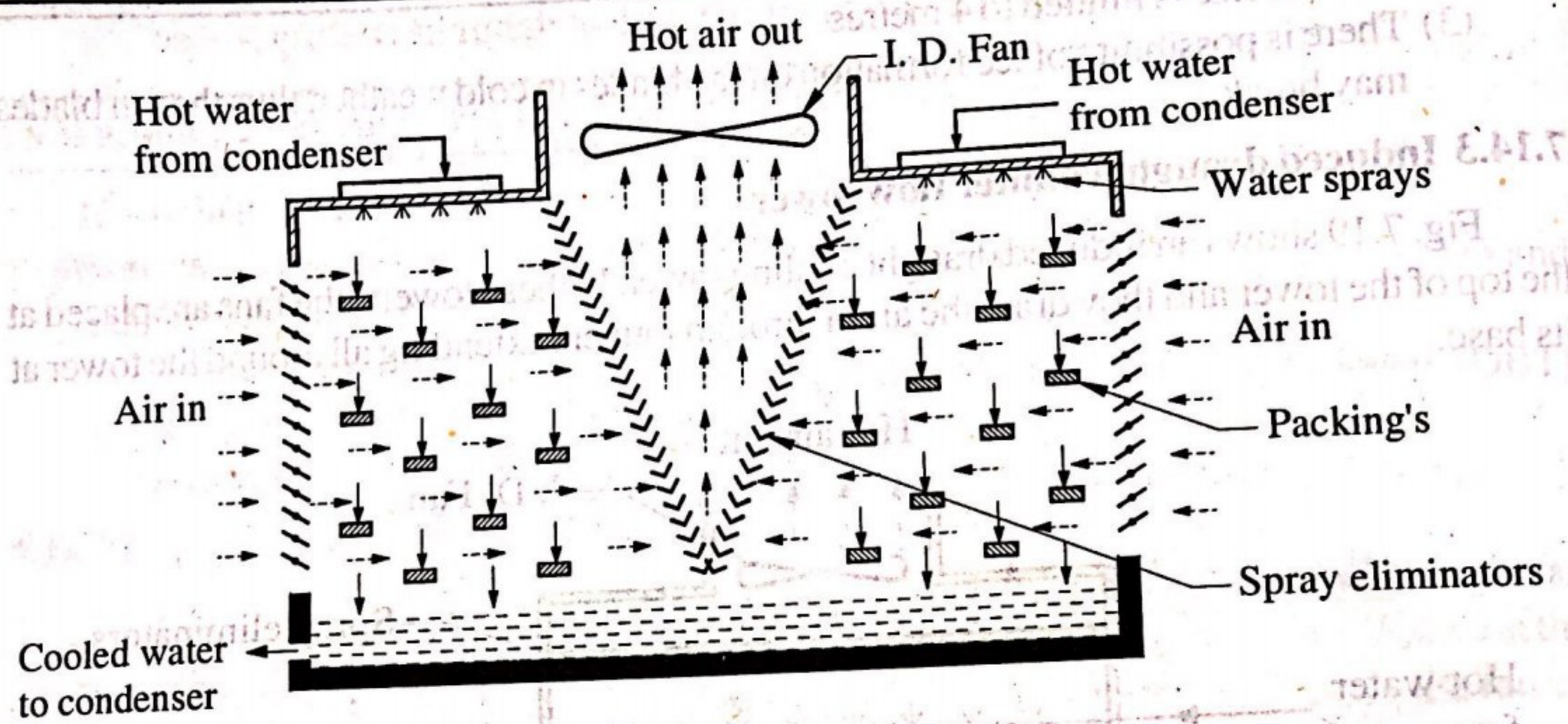


Fig. 7.20 Induced draught (cross-flow) cooling tower

- **Advantages of induced draught cooling towers :**

(1) The coldest water comes in contact with the driest air and warmest water comes in contact with the most humid air.
(2) In this tower, the recirculation of hot humid air is not a major problem.
(3) Larger fan size can be used.
(4) The power consumption per kg of water cooled is less compared with F.D. Fan system.
(5) Lower first cost due to the reduced pump capacity and smaller length of water pipes.
(6) This tower is capable of cooling through a wide range.

- **Disadvantages :**

(1) It is not economical up to the capacity of 15000 litres of water per minute. For larger quantity, the economics shifts in favour of the induced draft tower.
(2) Higher power motor is required to drive the fan compared to forced draught fan since it needs to handle hot air of high specific volume.
(3) Its maintenance cost is high.
(4) Fan bearings need to be cooled.

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7.14.5 Comparison between natural and mechanical draught cooling tower

Sr.No.	Principle	Natural draught cooling tower	Forced draught cooling tower
1.	Size and shape	It may be 125 m high and 100 m in diameter and has hyperbolic shape.	It may be 15 m high and thousands of meters in length and has long rectangular box shape.
2.	Initial cost	Considerably high	Nearly half of natural draught cooling tower
3.	Operating cost	Less since fans are not required	High since energy is required to run the fans.
4.	Maintenance cost.	Low because less no. of mechanical equipment is involved.	High because possibility of failure is more with this tower.
5.	Site suitability	Preferred when operating conditions couple low WBT and high relative humidity and for valley locations and sites of the plants near highways.	Preferred for all locations except the plants near highways and navigable waterways where fogging cannot be tolerated for safety reasons.
6.	Water loss	Less	More since fans create more pressure difference across the tower.
7.	Life	More	Comparatively less.

7.15 Cooling ponds

The cooling pond is one of the simplest method of cooling the condenser water. This method is less efficient than cooling tower.

The following type of cooling ponds are commonly used in practice :

(1) Natural and Directed flow cooling pond.

(2) Single deck and double deck cooling pond.

(3) Open and Louvre fence cooling pond.

(1) Natural and Directed flow cooling pond

In *natural flow* cooling pond, hot water coming out from the condenser is just allowed to flow into the pond as shown in Fig. 7.21. From hot water, heat is transferred to atmosphere by convection. This system is rarely used now a days.

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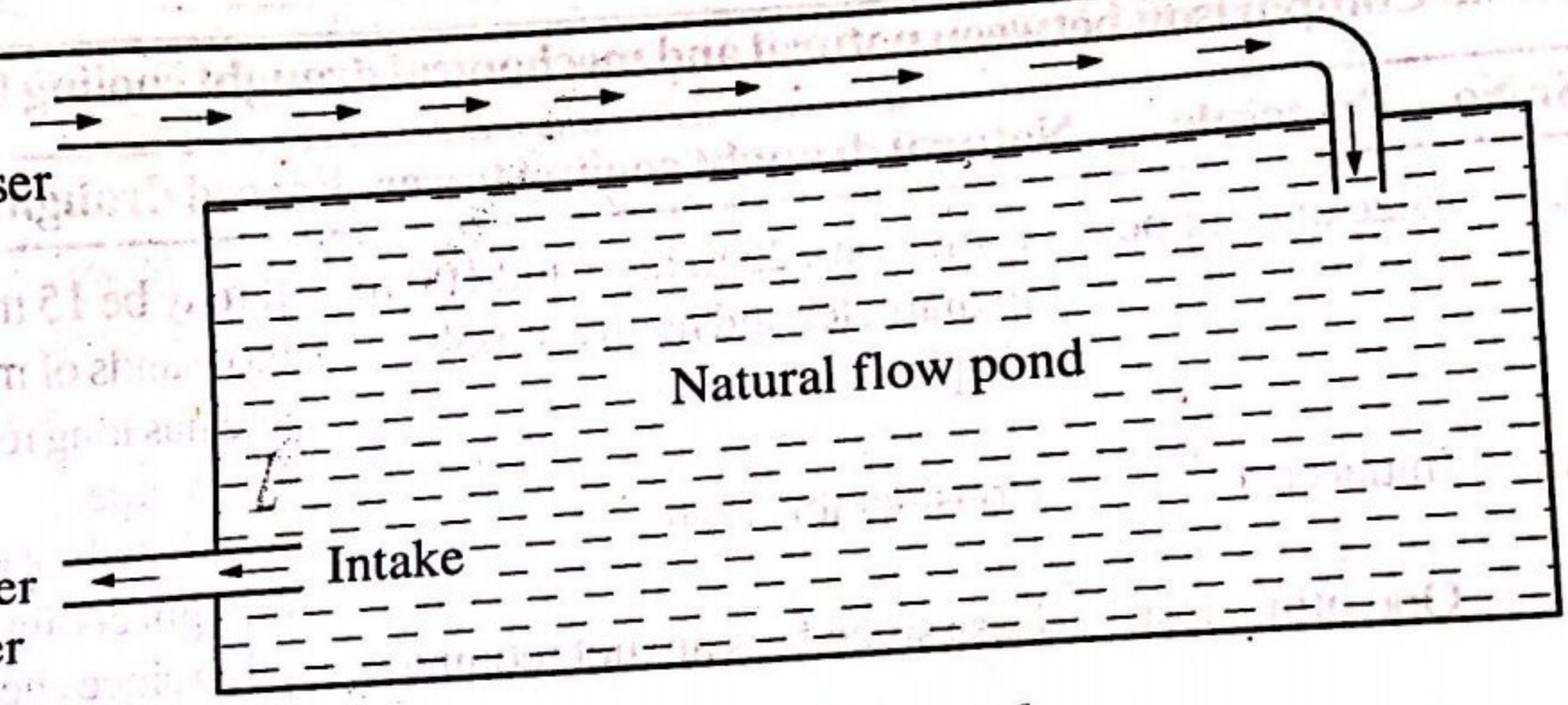


Fig. 7.21 Natural flow cooling pond

Fig. 7.22 shows ***directed flow*** cooling pond. Its cooling effect is more than the non-directed flow pond. The hot water coming out of condenser enters the middle channel and on reaching the far end divides into two currents, being directed by the baffle walls so as to traverse the pond several times before uniting at the intake point.

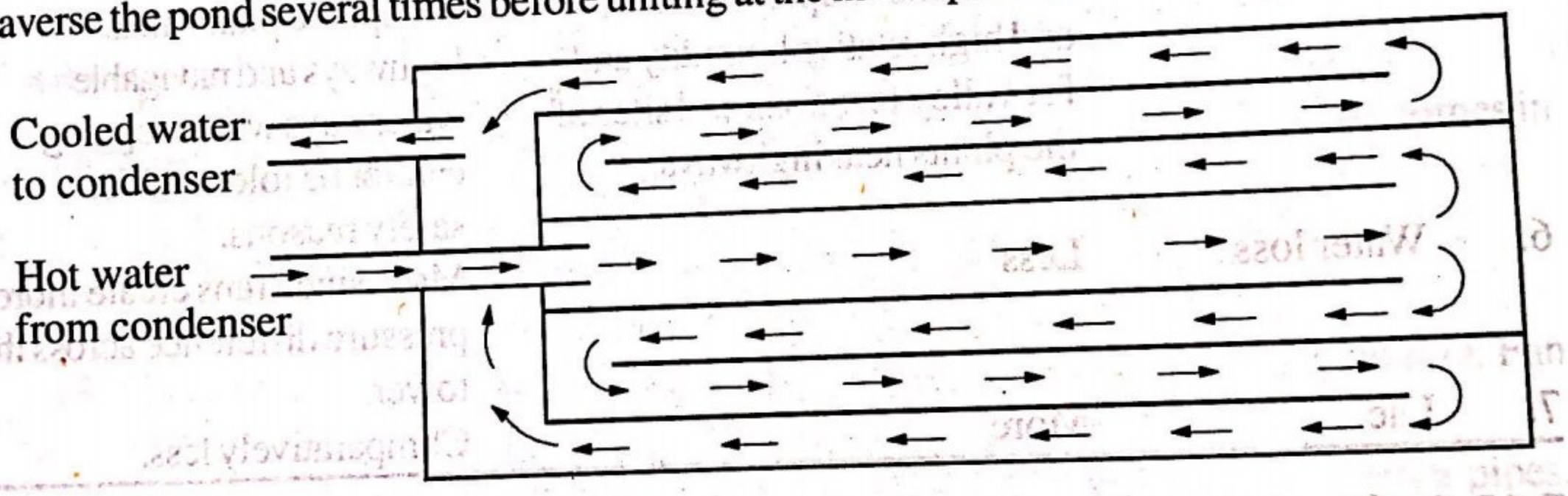


Fig. 7.22 Directed flow cooling pond

(2) Single deck and Double deck cooling pond

A single deck cooling pond is shown in Fig. 7.23. In this pond, spray nozzles are

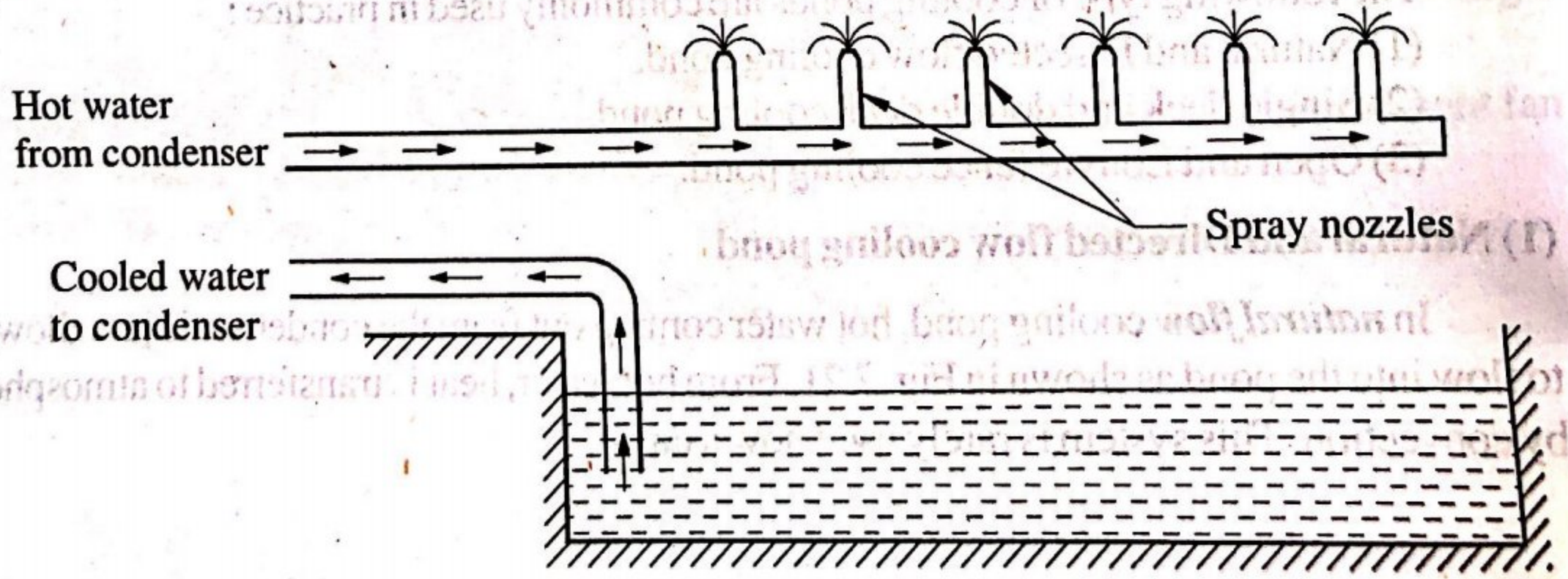


Fig. 7.22. Single deck spray pond

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arranged at the same elevation. In this system warm water received from the condenser is sprayed through the nozzles over a pond of large area and cooling effect is mainly due to evaporation from the surface of water.

In ***double deck*** spray pond, spray nozzle are arranged at different elevations as shown in Fig. 7.24. In this system cooling is more effective than single deck system.

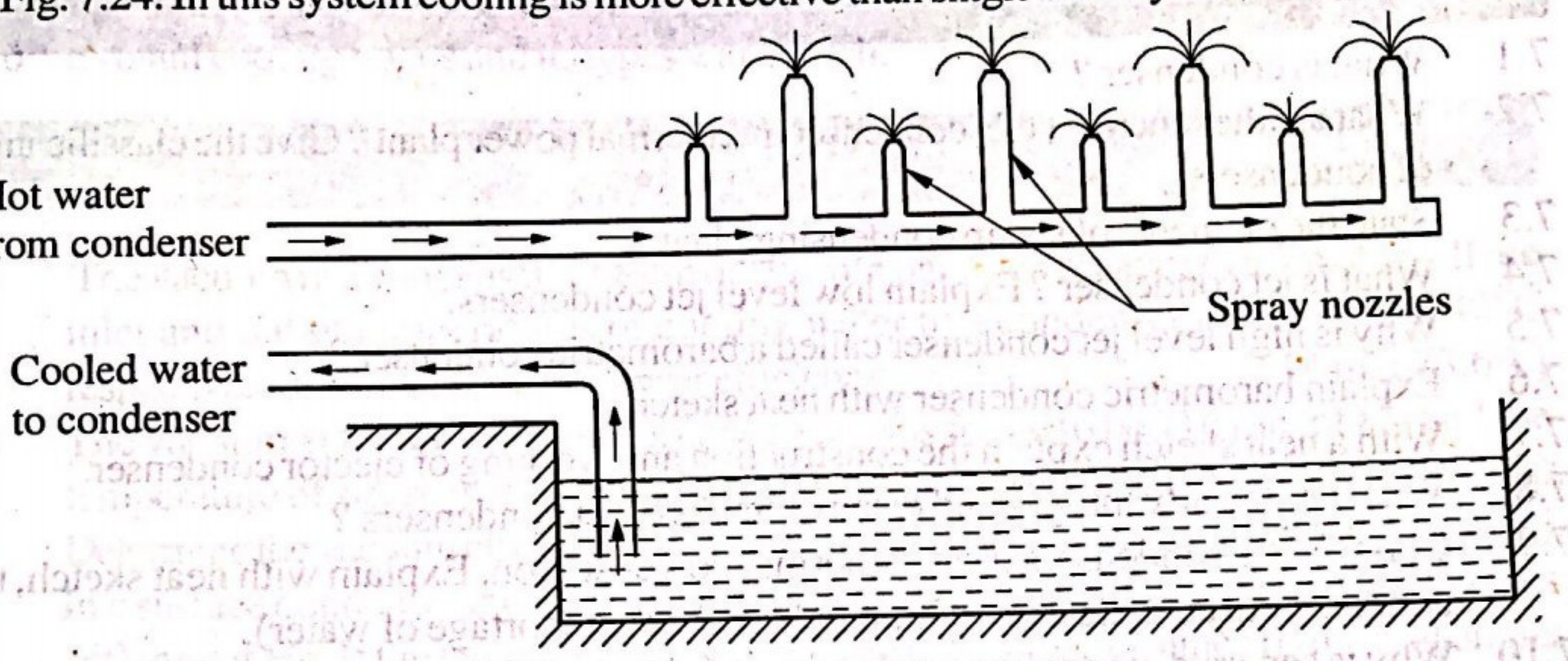


Fig. 7.24 Double deck spray pond

(3) Open and Louvre fence

The spray ponds discussed above (Single deck and Double deck) may be open type or with louvre fence. In case of open pond, the draft losses will be more if the wind velocity is high. This can be avoided by providing louvre fence as shown in Fig. 7.25.

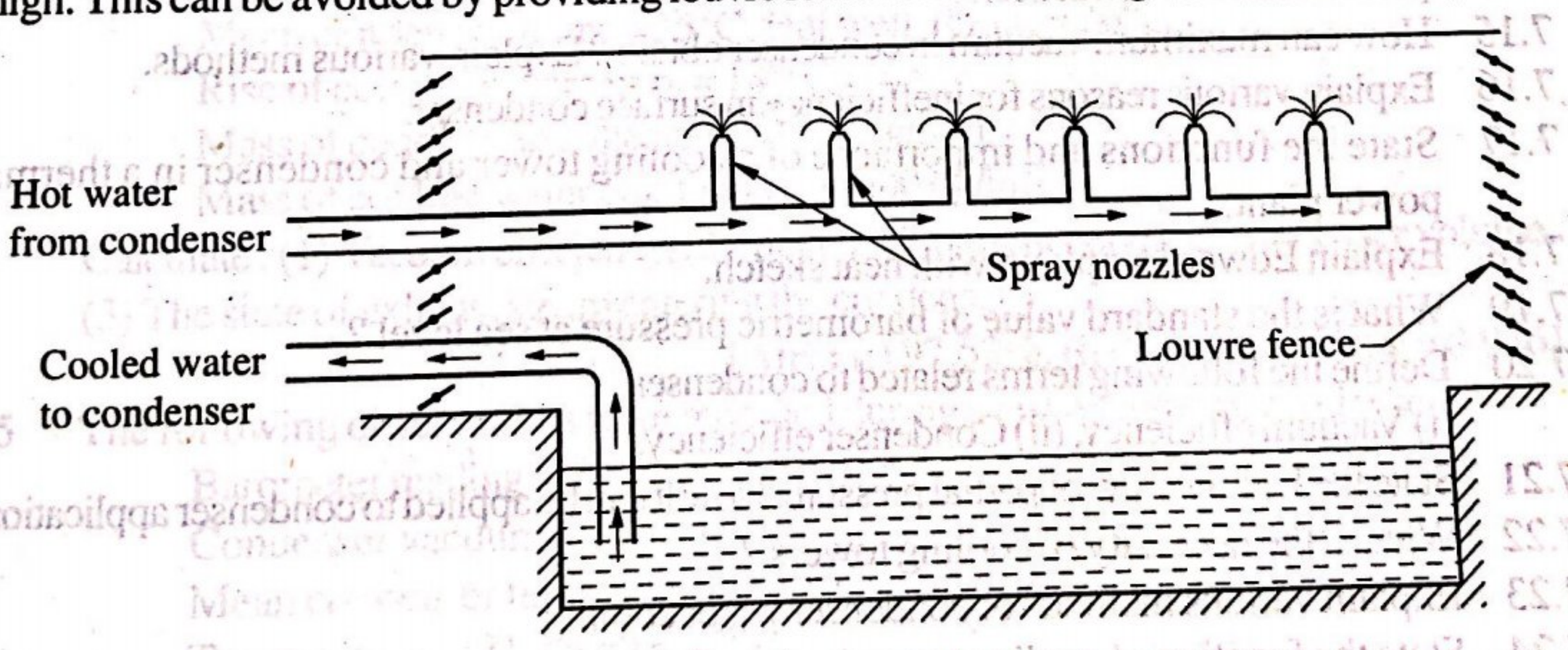


Fig. 7.25 Spray pond with louvre fence

- **Advantages of cooling ponds :**
  - (1) Its initial cost is less.
  - (2) Simple in design and arrangement.
- **Disadvantages :**
  - (1) It requires large area for cooling.

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(2) More amount of loss of water due to evaporation and windage run.
(3) Low cooling efficiency.
(4) There is no control over the temperature of cooled water.

Exercise

7.1 What is condenser ?
7.2 What are the functions of a condenser in a thermal power plant ? Give the classification of condensers.
7.3 State the elements of steam condensing plant.
7.4 What is jet condenser ? Explain low level jet condensers.
7.5 Why is high level jet condenser called a barometric condenser ?
7.6 Explain barometric condenser with neat sketch.
7.7 With a neat sketch explain the construction and working of ejector condenser.
7.8 What are the advantages and disadvantages of jet condensers ?
7.9 Classify the condensers used in thermal power station. Explain with neat sketch, the condenser used in desert location (where there is shortage of water).
7.10 Why is separate air cooling section provided in surface condensers ?
7.11 Explain double pass surface condenser with neat sketch.
7.12 State advantages and disadvantages of surface condenser.
7.13 Compare jet condenser with surface condenser.
7.14 What are the sources of air leakage in condenser ? and explain its effects on the performance of condensers.
7.15 How can maximum vacuum in condenser obtain ? Explain various methods.
7.16 Explain various reasons for inefficiency in surface condenser.
7.17 State the functions and importance of a cooling tower and condenser in a thermal power plant.
7.18 Explain Edward air pump with neat sketch.
7.19 What is the standard value of barometric pressure at sea level ?
7.20 Define the following terms related to condenser
(i) Vacuum efficiency, (ii) Condenser efficiency.
7.21 State the Dalton's law of partial pressure. How it can be applied to condenser application?
7.22 What is the necessity of cooling towers ?
7.23 Explain various methods of condenser water cooling.
7.24 State the function of cooling tower in a modern steam power plant. Describe with neat sketch the working of a mechanical type-cooling tower.
7.25 Classify different types of cooling tower used in power plant. Explain Natural draft cooling tower. Also explain the reason of its hyperbolic shape.
7.26 Explain the construction of natural draught cooling tower with diagram. Also write its advantages and limitations.

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7.27 Explain the working of mechanical cooling tower with the neat sketch. Also state its merits and demerits over natural cooling tower.

7.28 Why cooling towers are used in a power plant ? Explain the function of a cross flow cooling tower with neat sketch.

7.29 Compare mechanical draught cooling towers with natural draught cooling towers.

7.30 Explain cooling ponds and its types with sketch.

Unsolved Numerical Problems

7.1 The vacuum in a condenser is 68 cm of Hg with barometer reading 76 cm of Hg. If the inlet and outlet temperatures of cooling water to a condenser are 28°C and 42°C respectively. Calculate the condenser efficiency. **[Ans :** 73.6%]

7.2 The vacuum at the steam inlet to a condenser was found to be 716 mm of Hg and the temperature of steam in the condenser is 35°C, when barometer reads 760 mm of Hg. Determine the vacuum efficiency. **[Ans :** 99.74%]

7.3 In a surface condenser the vacuum maintained is 710 mm of Hg. The barometer reads 760 mm of Hg. If the temperature of condensate is 18°C, determine : (i) mass of air per kg of steam, (ii) vacuum efficiency of condenser. **[Ans :** 3.576 kg, 95.37 %]

7.4 In a surface condenser test, the following observation were recorded :

Condenser vacuum = 700 mm of Hg.
Barometer reading = 760 mm of Hg.
Mean condenser temp. = 35°C, Hot well Temp. = 30°C
Rise of cooling water temp. = 14°C
Mass of condensate collected = 16.75 kg/min.
Mass of cooling water circulated = 39600 kg/hr.

Calculate : (1) Vacuum efficiency, (2) Mass of air present per m^3 of condenser volume, (3) The state of exhaust steam entering the condenser.
[Ans : (i) 97.53%, (ii) 0.027 kg, (iii) $x = 0.946$]

7.5 The following observations were recorded during a test on a steam condenser :

Barometer reading = 765 mm Hg
Condenser vacuum = 710 mm Hg.
Mean condenser temperature = 35°C
Temperature of Hot well = 28°C
Condensate collected = 2000 kg/hr.
Cooling water circulated = 60,000 kg/hr.
Temperature of cooling water at inlet = 14°C
Temperature of cooling water at outlet = 30°C

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Determine : (i) vacuum corrected to standard barometric pressure of 760 mm Hg. (ii) quality of steam entering the condenser, (iii) condenser efficiency, (iv) mass of air present per kg of steam. [**Ans :** (i) 705 mm Hg, (ii) x = 0.818, (iii) 76.2%, (iv) 0.4 kg]

7.6 In a surface condenser test, the following observations were recorded :

Temperature of entering steam = 40°C
Dryness fraction of entering steam = 0.85
Condensate collected = 10 tons/h
Air leakage in condenser = 30 kg/h
Condensate temperature = 30°C

Determine : (i) the quantity of cooling water circulated, if the rise in temperature is 10°C, in kg/min, (ii) the capacity of the wet air pump in m^3/min.

[**Ans :** (i) 8310.29 kg/min. (ii) 838.73 m^3/h]

7.7 The air leakage into a surface condenser operating with a steam turbine is estimated as 86 kg/h. The vacuum near the inlet of air pump is 700 mm of Hg. when barometer reads 760 mm of Hg. The temperature at inlet of vacuum pump is 19°C. Determine : (i) the minimum capacity of the air pump in m^3/h, (ii) the dimensions of the reciprocating air pump to remove the air if it runs at 180 r.p.m. Take L/D = 1.5 and volumetric efficiency = 95%, and (iii) the mass of vapour extracted per minute.

[**Ans :** (i) 1246.91 m^3/h, (ii) D = 0.461 m, L = 0.691 m, (iii) 0.339 kg/min]

Objective Type Questions

1. A condenser in a steam turbine power plant
   (a) reduces back pressure of steam (b) increases expansion ratio of steam
   (c) reduces temperature of exhaust steam (d) all of the above
2. The following is not true for condenser
   (a) It increases thermal efficiency of the power plant.
   (b) It reduces corrosion of boiler tubes.
   (c) It increases the power plant size for given output.
   (d) It reduces amount of heat supplied per kg of steam.
3. The following is not a jet condenser.
   (a) evaporative condenser (b) barometric condenser
   (c) ejector condenser (d) all of the above.
4. In the following condenser, condensate extraction pump is required
   (a) low level jet condenser, (b) barometric condenser
   (c) ejector condenser (d) all of the above
5. A condenser where circulating water flows through tubes which are surrounded by steam.
   (a) jet condenser (b) surface condenser
   (c) barometric condenser (d) evaporative condenser

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6. The following is true for direct contact type condensers :
   (a) The power required to run the air pump is low.
   (b) The quantity of cooling water required per kg of steam condensed is less.
   (c) Their maintenance cost is high.
   (d) Vacuum maintain is higher.
7. _____ needs very small amount of circulating water and make up water.
   (a) surface condenser (b) ejector condenser.
   (c) barometric condenser (d) evaporative condenser.
8. The following is not a source of air leakage in surface condenser.
   (a) boiler feed water (b) condenser joints
   (c) cooling water (d) none of the above
9. The following is not the effect of air leakage in condenser.
   (a) reduces the thermal efficiency of plant.
   (b) increases the corrosive action.
   (c) increases the overall heat transfer from steam.
   (d) increases cooling water requirement.
10. The ratio of actual vacuum to the ideal vacuum in a condenser is called
   (a) vacuum efficiency (b) condenser efficiency
   (c) overall efficiency (d) expansion efficiency
11. The ratio of actual rise in the temperature of cooling water to the maximum possible rise in temperature of cooling water is known as
   (a) vacuum efficiency (b) condenser efficiency
   (c) overall efficiency (d) expansion efficiency
12. The actual vacuum in a condenser is equal to
   (a) gauge pressure + atmospheric pressure
   (b) gauge pressure – atmospheric pressure
   (c) barometric pressure + actual condenser pressure
   (d) barometric pressure – actual condenser pressure
13. The following is not applicable to hyperbolic cooling tower
   (a) It requires large ground area. (b) It's operating cost is less.
   (c) It's initial cost is considerably high. (d) It gives trouble free operation.
14. The following is not applicable to induced draught cooling tower.
   (a) Its maintenance cost is high.
   (b) It requires high power motor to run I.D. fan.
   (c) There is possibility of recirculation of hot, humid air.
   (d) The power consumption per kg of water cooled is less.

Answers

1. (d) **2.** (c) **3.** (a) **4.** (a) **5.** (b) **6.** (b) **7.** (d) **8.** (c) **9.** (c) **10.** (a)
11. (b) **12.** (d) **13.** (a) **14.** (c)

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GTU Study Materials

Chapter 8
Feed water treatment

8.1 Necessity of feed water treatment
8.2 Different impurities found in feed water
8.3 Effect of impurities
8.4 pH and its role in corrosion and scale formation
8.5 Feed water treatment
8.6 Internal Water treatment systems
8.7 External water treatment systems
8.8 Hot lime soda process
8.9 Zeolite or Ion exchange process
8.10 Demineralization plant
8.11 Reverse Osmosis process
8.12 Sea water treatment using reverse osmosis
8.13 De-aeration

8.1 Necessity of feed water treatment

Steam turbine power plants lose a fraction of their water and steam during operation due to the leakage from glands, fittings and bearings, escape with non-condensable gases in deaeration process, boiler blowdown and other causes. *This loss amounts to be about 1 to 3 percent of total flow*. Since this loss of water and steam is a continuous process and so this loss must be made up continuously during operation. It is highly essential that this make-up water must be treated well in water treatment plants to maintain water and steam purity. Raw water contains a variety of impurities, such as suspended solids and turbidity, organics, hardness (salts of calcium and magnesium), alkalinity (bicarbonates, carbonates, hydrates), other dissolved ions (sodium, sulphate, chloride, etc), silica, and dissolved gas (O_2, CO_2).

These impurities deposit on heat transfer surfaces in a power plant. These materials like dirt and scale have low conductivity and therefore impede heat transfer severely. Deposits of scales forming impurities on boiler tubes will restrict the water flow also. It can also raise the metal temperature with resultant tube rupture. The other effects of these solids are the corrosion and erosion of boiler tubes, turbine blades, and condenser tubes. Therefore, it is absolutely necessary to have a separate water softening plant to polish the water taken from outside source (well, lake or river) to counterbalance the losses in the system. The general outline of the feed water supply system is shown in Fig. 8.1.

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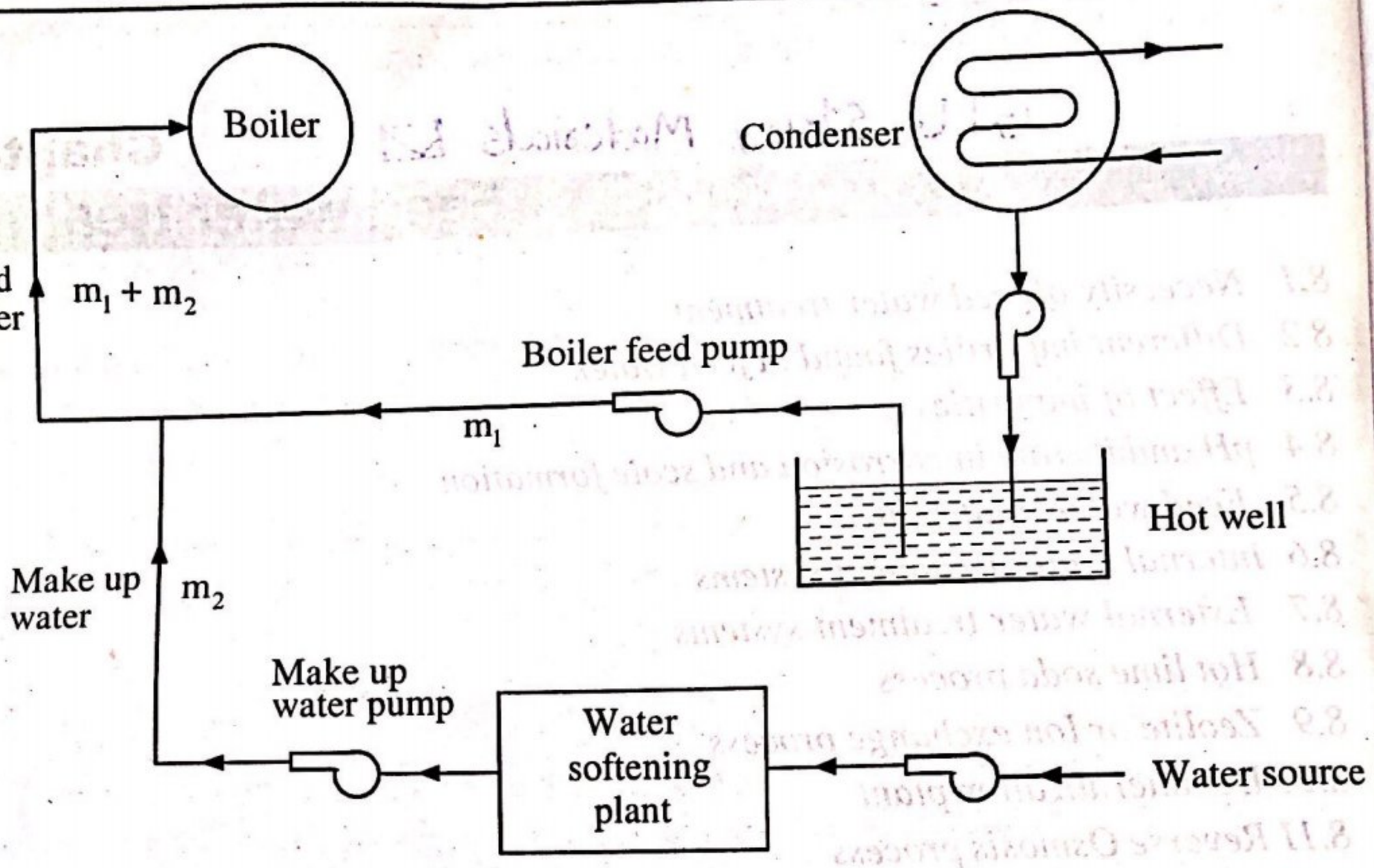


Fig. 8.1 Position of water softening plant

8.2 Different impurities found in feed water [Dec. '13, May '13, Dec. '11]

The water contains impurities and, therefore, those must be removed before feeding to the boiler. The impurities present in the feed water can be classified as given below.

(1) Undissolved and suspended materials

(a) Turbidity and sediment :

Turbidity in water is suspended solid particles in soluble matter like mud, sediment, sand, etc. that settle rapidly on standing. Their presence is undesirable because heating or evaporation produces hard stony scale deposits on the heating surface and clog fluid system. These impurities can be removed by filtration and settling coagulation. The turbidity of boiler feed water should not exceed 5 ppm (A standard of measurement of hardness is taken as being the amount of calcium carbonate ($CaCO_3$) in the water and is referred to in part per million (ppm) or grains per gallon $\times$ 17.1 = ppm).

(b) Sodium and Potassium salts :

Sodium and Potassium salts area alkaline in nature and accelerate the corrosion. They are extremely soluble in water and do not deposit unless highly concentrated.

(c) Chlorides : They cause increased corrosive action of water.

(d) Iron :

Generally ground surface water contains less than 5 ppm but even 0.3 ppm can create trouble in the feed water system by soft scale formation and accelerating the corrosion. Ferrous bicarbonate is most common soluble iron in water, it deposits becomes yellowish and reddish sediment of ferric hydroxide if exposed to air.

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(e) Manganese : Its presence is equally troublesome as iron.

(f) Silica :

Its presence in water is highly objectionable as it forms very hard scale in boiler and forms insoluble deposits on turbine blades. Its presence in feed water is reduced as low as 10-50 parts per billion (ppb). Most natural water contains Silica ranging from 1 to 100 ppm.

(g) Microbiological growths :

The microbiological growths in surface water (river and lake) include bacterial slimes, molds, diatons, algae, etc. These can cause coating on heat exchanger and clog the flow passages and reduce the heat transfer rates.

(h) Colour :

Its presence in water causes foaming in boilers and may interfere with treatment processes. It is generally removed by chlorination or absorption by activated carbon.

(2) Dissolved salts and minerals

(a) Calcium and Magnesium salts :

The presence of these salts is recognised by the hardness of the water. The hardness of water is tested by standard soap test. These salts are present in the water in the form of carbonates, bicarbonates, sulphates and chlorides. Water having carbonate hardness may be softened by sufficient heating or boiling. CO_2 is liberated and relatively insoluble precipitate of $CaCO_3$ and $MgCO_3$ are formed which can be removed by simple blow down method. Hardness of water is caused by the presence of chlorides, sulphates and nitrates of calcium and magnesium cannot be removed by just boiling because they form a hard scale on heating surfaces.

(3) Dissolved gases

(a) Oxygen :

Oxygen is corrosive to iron, zinc, brass and other metals. It is present in surface water in dissolved form. It causes corrosion and pitting of all metals parts with which it comes in contact especially at high temperatures.

(b) Carbon dioxide:

Carbon dioxide produces corrosion and scale forming compounds by combining with other materials. Majority of water contains 2 to 50 ppm of CO_2.

(4) Other materials

(a) Free mineral acid :

Usually they present in water as sulphuric or hydrochloric acid. Their presence in water is always undesirable as it may result in the corrosion. The presence is reduced by neutralization with alkalies.

(b) Oil :

Generally the lubricating oil is carried with steam into the condenser and through the feed water system to the boiler. It causes sludge, scale and foaming in boilers. It is generally removed by strainers and baffle separators.

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8.3 Effect of impurities

The following troubles arise due to one or more of the water impurities discussed above.

(1) Scale formation

The concentration of dissolved salts in hard water increases progressively due to continuous evaporation of water in boilers and finally these salts are precipitated on the inner walls of the boiler. If the precipitated matter forms a hard adhering coating inside the boiler surface, they are called as scales. The mechanism of scale formation is shown in Fig. 8.2.

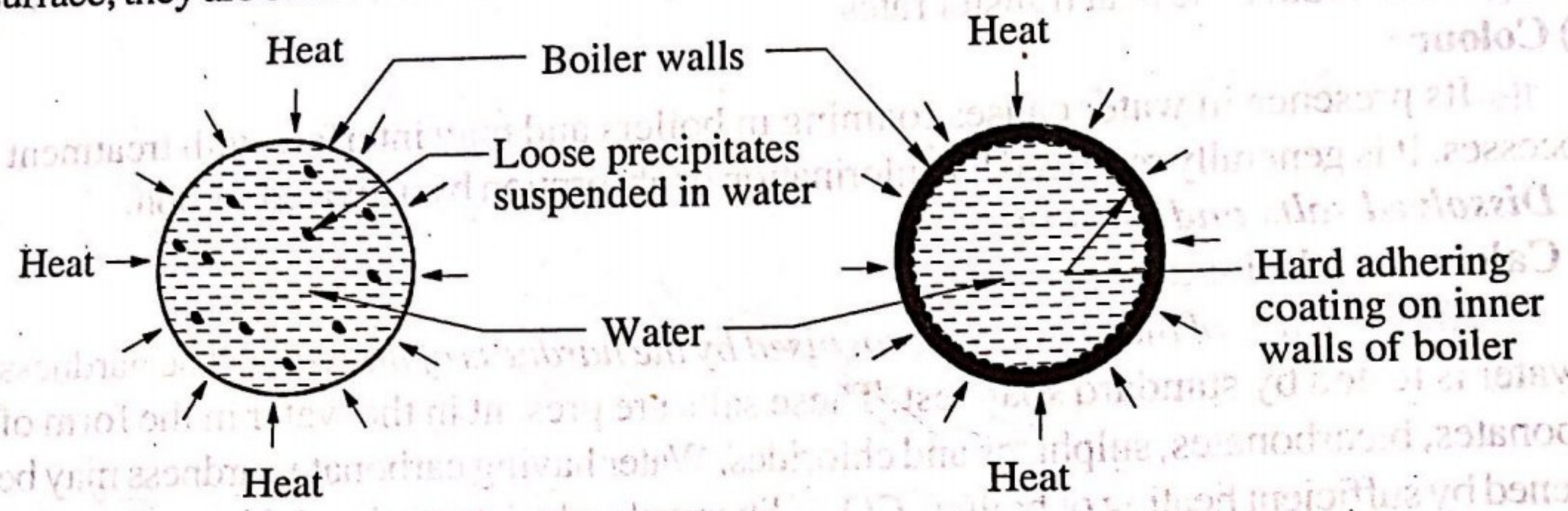


Fig. 8.2 Scale formation

The formation of scale reduces heat transfer and simultaneously raises the temperature of the metal wall. When temperature of boiler tube material reaches 450°C to 500°C, there is a serious danger of overheating and consequently rupture of boiler tubes. The scaling results from the decrease of solubility of some salt with increase of temperature. Scale is mainly due to the salts of calcium and magnesium as also to a certain extent the salts of silicates.

Calcium and magnesium bicarbonates are broken down by moderate heating (100°C) into relatively insoluble monocarbonate and CO_2 as given by the following chemical reactions.

$$Ca(HCO_3)_2 + \xrightarrow[100°C]{Heat} CaCO_3\,(\text{soft sludge})\downarrow + CO_2\uparrow + H_2O$$

$$Mg(HCO_3)_2 + \xrightarrow[100°C]{Heat} MgCO_3\,(\text{soft sludge})\downarrow + CO_2\uparrow + H_2O$$

The less soluble salts such as calcium carbonate precipitate and settle in the boiler. The CO_2 formed is carried with steam.

The scale chokes the flow in the piping system and thus requires increased pressure to maintain water delivery. The accumulation of scale may become so thick that the heat transfer rate drop from flue gas to water. This will result in overheating to metal side, blistering and rupturing. When scale is formed, tubes are cleaned with electric-powered rotary brushes and cutters are pushed through the tubes during boiler overhaul.

The tough and tightly adhered scales are worst from the point of removal. The formation of scale is prevented either by removing or reducing the contents of calcium, magnesium or silica before feed water reaches the boiler or by internal treatment.

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(2) Corrosion

The corrosion is the eating away process of boiler metal. It produces pits, grooves, and cracks or a general wastage of material. Eventually, this calls for major repairs or expensive shutdowns or replacement.

The corrosion of boilers, economisers, feed water heaters, and piping is caused by an acid or low pH in addition to the presence of dissolved oxygen and CO_2 in the boiler feed water. The presence of oxygen is mostly responsible for corrosion among all other factors. Generally it should not exceed 0.5 CC per litre. The carbon dioxide is next to oxygen which is responsible for corrosion, it comes out of bicarbonates on heating and it combines with water to form weak acid known as carbonic acid. This acid slowly reacts with iron and other metals to form their bicarbonates. The new bicarbonates of metals formed are decomposed by heat once more and carbon dioxide is again liberated. This gas again unites with water to form carbonic acid and the cycle is repeated.

The corrosion can be controlled by adding alkali salts to neutralise acids in water and raise the pH value. The effect of CO_2 is neutralised by the addition of ammonia for neutralising amines in water. This is necessary because CO_2 lowers the pH of the boiler feed water. The effect of oxygen is reduced only by removing the oxygen from water. The corrosion of metal surfaces can be prevented by applying protective coating of amines to the internal surfaces of boilers and economisers.

(3) Priming, Foaming and carryover : **[June '11]**

A layer of ***foam*** in the boiler drum is caused by soluble and insoluble salts and other organic impurities which are carried in suspension. Foaming prevents the free escape of steam bubbles as they rise to the surface of water. Oil and other impurities which may be present in boiler water may cause foaming. Foaming can be avoided by adding castor oil and antifoaming chemicals and by adding sodium aluminate ($NaAlO_2$).

Priming is the passing of small water particles with steam as it leaves the boiler caused by impurities in water. Thus, the steam generated is wet. ***Carry over*** refers to passing of dissolved solids with steam while it leaves the boiler. The priming effect is reduced by installing steam purifiers, lowering water level in the boiler drum and maintaining constant load on the boiler.

The processes of priming and foaming usually occur together. The effect of priming, foaming or carryover is to carry the water which contains dissolved salts or undissolved slugs with the steam. The carry over of boiler water solids disrupts operation of the equipments coming in contact with the steam. The deposits are formed in the steam piping, valves, superheaters and turbine blades. These deposits erode the turbine blades, superheater tubes and interfere with lubrication system.

(4) Caustic Embrittlement :

The caustic embrittlement is caused by using highly alkaline water (caustic in water).

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It causes the brittleness of metal and as a result cracks appear in boiler shell below water level. This is caused by long exposure of boiler steel to a combination of stress and highly alkaline water.

The following three conditions contribute to embrittlement.

(i) When the boiler water contains free hydroxide alkalinity and some silica, it has been always found that the feed water was high in sodium-bicarbonate which broke down into sodium carbonate in the boiler and partially hydrolized as shown by the following reaction in case of embrittlement.

$$Na_2CO_3 + HOH \rightarrow CO_2 \uparrow + 2NaOH$$

(ii) Slow leakage of boiler water through a joint or seam.

(iii) Boiler metal is highly stressed at the point of leakage. This may be caused by faulty riveting, misalignment and expansion.

This defect is not very common. However, it must be considered in every water treatment plant and measures must be taken to prevent it.

Embrittlement can be controlled by

(i) eliminating free sodium hydroxide from boiler water.

(ii) maintaining a definite ratio of sodium nitrate to sodium hydroxide.

(iii) using waste sulphite liquor.

8.4 pH and its role in corrosion and scale formation [Dec. '11]

pH value of water is the logarithm of the reciprocal of hydrogen ion concentration. It is number from 0 to 14 with 7 indicating neutral water. The pH is a number denoting the degree of acidity or alkalinity of a substance. It is derived by measuring the amount of hydrogen ion (H^+) in grams per litre of solution. For example, pure water ionizes to produce 10^{-7} grams of H_2 ions per litre.

$$\therefore \text{pH value of water} = \log\left(\frac{1}{10^{-7}}\right) = 7 \text{ (at } 20^{\circ}\text{C temperature)}$$

Therefore, pure water being neutral solution, any solution producing more hydrogen iron than pure water will be acidic and degree is governed by the difference. A sample of water having pH value less than 7 indicates the acidity of water and pH value more than 7 indicates the alkalinity of water.

If a sample of water contains $1\,H^+$ ion for every $1\,OH^-$ ion, they balance other and they combine to form neutral water.

(1) Role of pH in corrosion :

According to electro-chemical theory all metals also tend to dissolve in water to a certain extent. Water in contact with iron dissolves very small quantity of iron into solution. The water quickly becomes saturated with iron which stops further dissolving.

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Fe (metallic state) = Fe^{++} (ferrous ion when goes into solution)

Water in natural condition also ionizes to a very small extent and forms positive H_2 ions and negative hydroxyl ions (OH^-) as given by the following equation :-

$$H_2O \rightarrow H^+ + OH^-$$

The presence of excess hydrogen (H^+) ions makes the water acidic and corrosive while the presence of excessive hydroxyl (OH^-) ions makes the water alkaline.

The ferrous ions combine with negatively charged hydroxyl ions to form a ferrous hydroxide.

$$Fe^{++} + 2OH^- \rightarrow Fe(OH)_2$$

The dissolved oxygen of water oxidizes formed ferrous hydroxide to ferric hydroxide as given by the following equation

$$4Fe(OH_2) + O_2 + 2H_2O = 4Fe(OH)_3 \downarrow \text{(Insoluable)}$$

The formed ferric hydroxide is much less soluble than the ferrous hydroxide and tends to precipitate. This in turn allows more ions to go into the solution.

Another important reaction given below in which hydroxyl ions from the water unite with the ferrous ions and hydrogen ions are left free in the water.

$$Fe^{++} + 2HOH\,(\text{water}) = Fe(OH)_2 + 2H^+$$

These excess hydrogen ions form a protective coating over the metallic surface and tend to prevent further solution of metallic ion. The dissolved oxygen in the water also tends to combine with free excess hydrogen ions and forms water again as given by the following equation.

$$4H^+ + O_2 = 2H_2O$$

Therefore, the dissolved oxygen also removes the protective layer of hydrogen ions and allows more iron to dissolve in the water.

Electro-chemical theory of corrosion as explained above states that the corrosion is controlled by the free hydrogen ions present in the water. The acidic or alkaline nature of the water is also responsible for the corrosion.

The role of pH in corrosion of metals is extremely important. The corrosion rate of iron in the absence of oxygen is proportional to pH upto a value of 9.6. At this point, hydrogen gas formation and dissolving of iron practically stop. This is the same pH produced by a saturated solution of ferrous hydroxide $Fe(OH)_2$.

The oxygen in the water unites with the ferrous hydroxide to form ferric hydroxide.

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This reaction lowers the pH of the solution and tends to stimulate corrosion.

Alkalinity adjustment and film formation are closely related. The pH value of feed water should be maintained greater than 9.6 to reduce the corrosion effects caused by the reason mentioned above. The required alkalinity of feed water is adjusted by adding soda ash, caustic soda or trisodium phosphate.

(2) Role of pH in scale formation :

The calcium carbonate is one of the most troublesome deposits responsible for scale formation. The calcium hardness alkalinity and pH are inter-related variables in scale formation.

As shown in Fig.8.3, water with 700 ppm calcium hardness and 100 ppm alkalinity at 7 pH lies at point A on the curve. The calcium carbonate will not precipitate being point A on the curve. If the water at the point B having 700 ppm calcium hardness and 130 ppm alkalinity is considered, the water is not in equilibrium being the point above the curve, the calcium carbonate will precipitate. The area below the curve indicates a tendency for $CaCO_3$ to dissolve. Water at point B can be brought back on the curve by reducing the hardness or alkalinity as shown in Fig. 8.3. Therefore, waters with the same calcium hardness may have different scale forming tendencies.

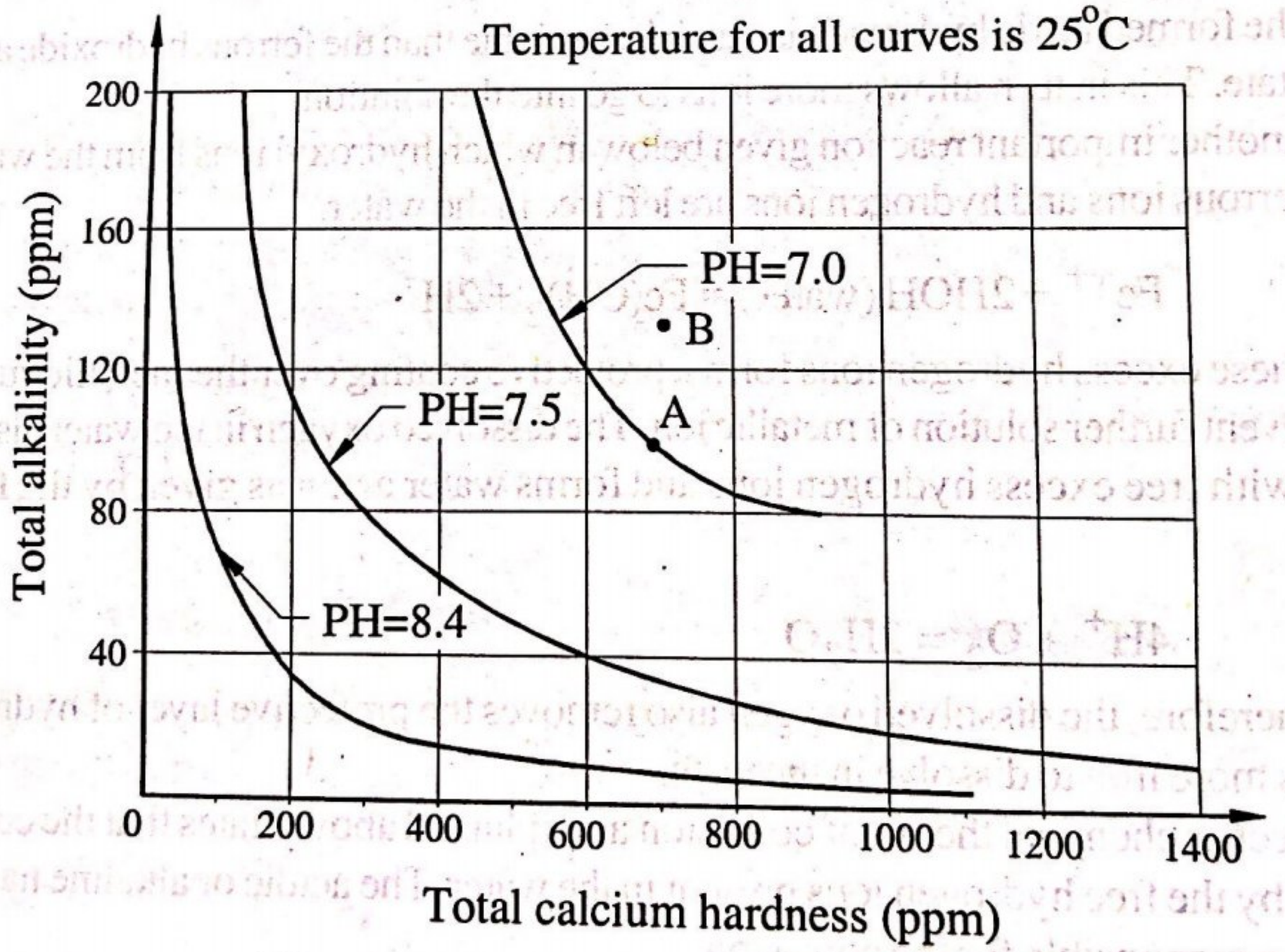


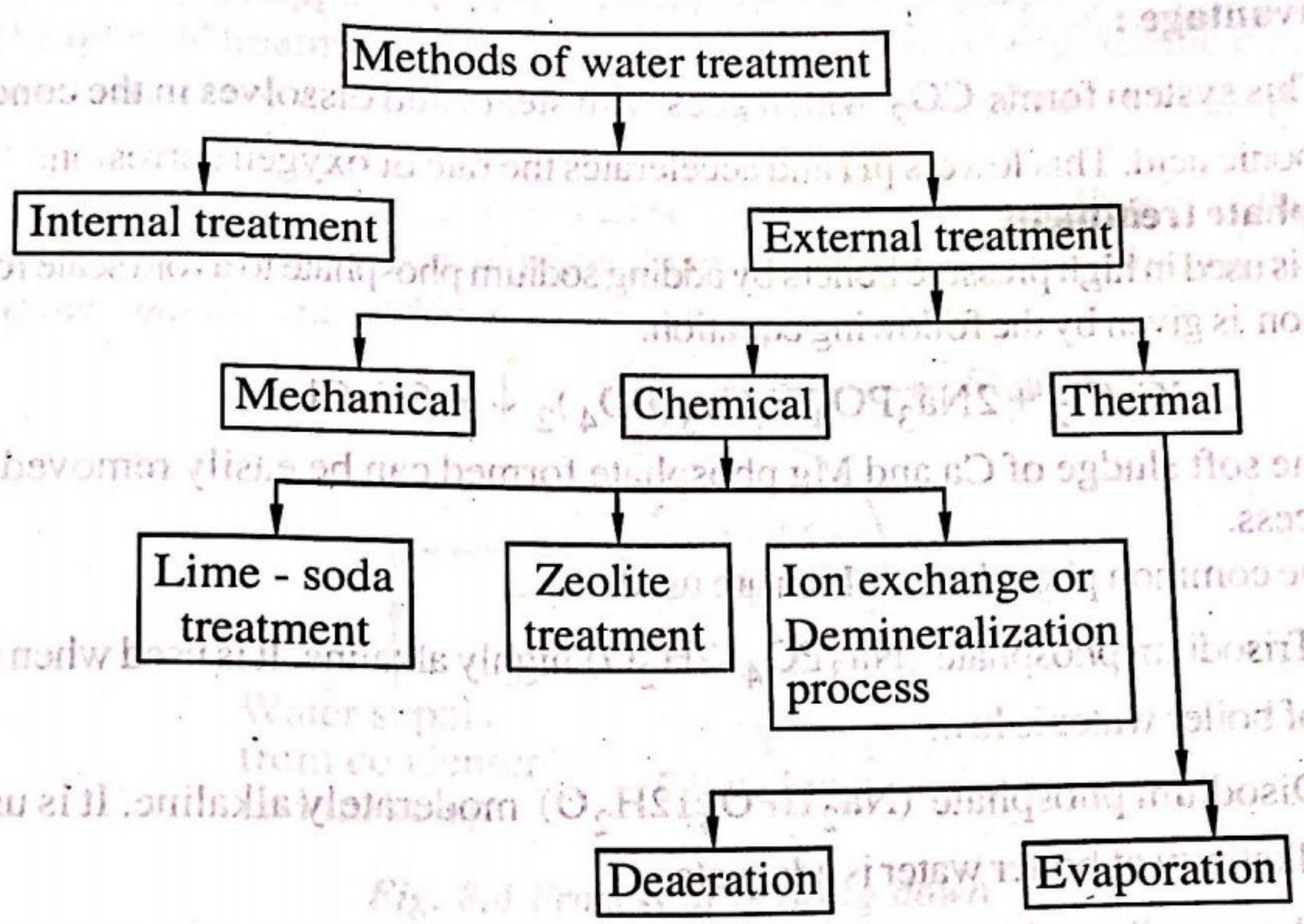
Fig. 8.3 Role of pH in scale formation

8.5 Feed water treatment

The main objective of the feed water treatment is to remove the suspended solids, dissolved solids and dissolved gases from water before supplying to the boiler. Various methods of water treatment are given below

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8.6. Internal Water treatment systems

The internal feed water treatment is accomplished by adding chemicals to boiler water. *The dissolved solids in the water are removed in the boiler itself by a chemical treatment then the method is known as* ***internal treatment***. It is carried out to prevent scale formation and to nullify the effects of external treatment. Some important methods of internal treatment are as follows :

(1) Sodium carbonate (Soda ash) treatment :

In low pressure boilers, scale formation can be avoided by addition of Na_2CO_3 (sodium carbonate) to boiler. The added sodium carbonate reacts with sulphates of calcium and magnesium in boiler water to form calcium and magnesium carbonates.

$$CaSO_4 + Na_2CO_3 = CaCO_3 \downarrow + Na_2SO_4$$

At high temperature, sodium carbonate reacts with water to form some free caustic soda (NaOH) as given by the following equation.

$$Na_2CO_3 + 2H_2O + Heat = NaOH + H_2O + CO_2$$

The free caustic soda formed reacts with the soluble magnesium salts to form insoluble magnesium hydroxide sludge as given below :

$$MgSO_4 + 2NaOH = Mg(OH)_2 \downarrow + Na_2SO_4$$

This system of cleaning destroys sulphate hardness.

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- **Disadvantage :**

This system forms CO_2 which goes with steam and dissolves in the condensate t form carbonic acid. This lowers pH and accelerates the rate of oxygen corrosion.

(2) Phosphate treatment

It is used in high pressure boilers by adding sodium phosphate to avoid scale formation. The reaction is given by the following equation.

$$3CaCl_2 + 2Na_3PO_4 = Ca_3(PO_4)_2 \downarrow + 6NaCl$$

The soft sludge of Ca and Mg phosphate formed can be easily removed by blow down process.

The common phosphates which are used.

(i) Trisodium phosphate ($Na_3PO_4 12H_2O$) highly alkaline. It is used when alkalinity of boiler water is low.

(ii) Disodium phosphate ($Na_2HPO_4 12H_2O$) moderately alkaline. It is used when alkalinity of boiler water is adequate.

(iii) Mono-sodium phosphate (NaH_2PO_4) slightly acidic. It is used when alkalinity of boiler water is too high.

(3) Sodium aluminate ($NaAlO_2$) treatment :

The scale formation can be avoided by addition of $NaAlO_2$ to boiler.

$$NaAlO_2 + 2H_2O = NaOH + Al(OH)_3 \downarrow$$

NaOH reacts with magnesium salts to form sludge

$$MgCl_2 + 2NaOH = Mg(OH)_2 \downarrow + 2NaCl$$

$Al(OH)_3$ and $Mg(OH)_2$ produced inside the boiler can be removed by blow down operation.

(4) Colloidal treatment :

The sludge formed either with carbonate system or phosphate system is sometimes objectionable, therefore organic compounds are added in the feed water to keep the sludge in circulation until it can be removed through blowdown.

It is used in low pressure boilers to prevent scale formation by adding organic substances like tanin, lignins, starch, kerosene etc. These substances get coated over scale forming precipitates. It yield non-sticky and loose deposits similar to sludge which can be removed by blow down process.

(5) Blow down system

Water entering the boiler may contain some dissolved solids. The concentration of these solids goes on increasing as the water is vaporised. Beyond a certain limit of concentration, these solids may cause foaming and priming. The upper limit is 2500 to 5000 ppm for low

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pressure boilers and 500 ppm or even less for high pressure boilers.

The internal treatment precipitates undesirable salts in most desirable form. This removes the hardness of water but produces equivalent amount of sludge that is potentially troublesome.

The concentration of these solids can be reduced by drawing off some of the quantity of the boiler water from the bottom of boiler drum as shown in Fig. 8.4. *This process is called blowing down and discharged water is known as blow down.*

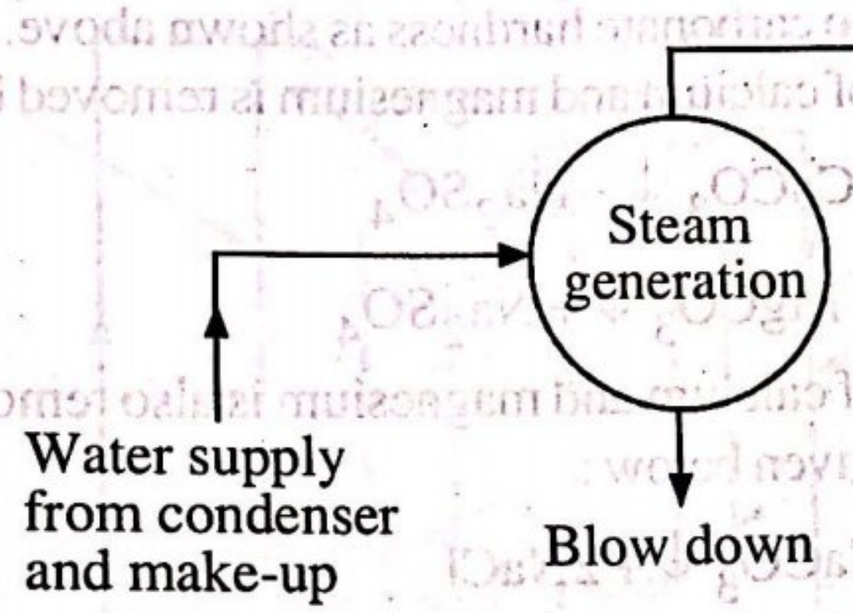


Fig. 8.4 Process of blowing down

The blow down water mainly contains the undesirable impurities which concentrate at the bottom of drum. As a result of blow down the concentration of these impurities inside the boiler drum can be temperarily reduced. Therefore a boiler may have periodical blow down so that the concentration of impurities can be kept within permissible limits.

8.7 External water treatment systems

The external water treatment becomes essential when the make-up water quantity is large and contains considerable suspended and dissolved solid material. The suspended solid material is generally removed by mechanical means. The dissolved solids are removed with the use of chemical treatment and dissolved gases are generally removed by thermal treatment.

Mechanical treatment includes sedimentation, coagulation and filtration. Suspended matter can be removed easily by these processes. *Sedimentation involves allowing the water to stand quietly for some time*. In this way the solid matter settles down and is removed periodically. *In case of coagulation some coagulatants like aluminium sulphate, sodium aluminate or ferrous sulphate are added to the impure water*. This removes the minute colloidal suspensions. ***Filtration*** consists in passing the water through filters. The suspended matter adheres to the filter material. The filters may be either gravity filter or pressure filter.

Chemical treatment includes hot lime soda process, zeolite ion exchange process and demineralization. ***Thermal treatment*** includes distillation and deaerative heating of water.

8.8 Hot lime soda process

In this method, lime $Ca(OH)_2$ (calcium hydroxide) and soda ash Na_2CO_3 (sodium carbonate) use as treating elements. These treating elements react with all types of calcium and

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magnesium salts and precipitate them. These reactions are generally carried out at the boiling point (100°C) of the water where the water is heated with the help of the steam.

The chemical reactions involved in lime soda processes are :

$$Ca(HCO_3)_2 + Ca(OH)_2 = 2CaCO_3 \downarrow + 2H_2O \quad ...(8.1)$$

$$Mg(HCO_3)_2 + 2Ca(OH)_2 = 2CaCO_3 \downarrow + Mg(OH)_2 \downarrow + 2H_2O \quad ...(8.2)$$

The lime precipitates the carbonate hardness as shown above.

The sulphate hardness of calcium and magnesium is removed by soda ash.

$$CaSO_4 + Na_2CO_3 = CaCO_3 \downarrow + Na_2SO_4 \quad ...(8.3)$$

$$MgSO_4 + Na_2CO_3 = MgCO_3 \downarrow + Na_2SO_4 \quad ...(8.4)$$

The chloride hardness of calcium and magnesium is also removed by soda and lime respectively. The equations are given below :

$$CaCl_2 + Na_2CO_3 = CaCO_3 \downarrow + 2NaCl \quad ...(8.5)$$

$$MgCl_2 + Ca(OH)_2 = Mg(OH)_2 \downarrow + CaCl_2 \quad ...(8.6)$$

The magnesium carbonate formed as shown by the equation (8.4) further reacts with lime and forms magnesium hydroxide sludge. The chemical reaction is given below.

$$MgCO_3 + Ca(OH)_2 = Mg(OH)_2 + CaCO_3 \downarrow \quad ...(8.7)$$

The magnesium hydroxide ($Mg(OH)_2$) acts as a coagulant and has the property of absorbing soluble silica from solution.

Trisodium phosphate and caustic soda are sometimes used in addition to lime and soda to precipitate and remove calcium and magnesium hardness very effectively. The reactions are as follows :

$$3Ca(HCO_3)_2 + 6NaOH = 3CaCO_3 \downarrow + 3Na_2CO_3 + 6H_2O \quad ...(8.8)$$

$$3CaCO_3 + 2Na_3PO_4 = Ca_3(PO_4)_2 \downarrow + 2Na_2CO_3 \quad ...(8.9)$$

$$Mg(HCO_3)_2 + 4NaOH = Mg(OH)_2 \downarrow + 2Na_2CO_3 + 2H_2 \quad ...(8.10)$$

$$3MgSO_4 + 2Na_2PO_4 = Mg_3(PO_4)_2 \downarrow + 3Na_2SO_4 \quad ...(8.11)$$

The arrangement of hot lime-soda process with filtration is shown in Fig. 8.5. The raw make-up water enters through a float controlled regulating valve and then passes over vent condenser and then sprayed into the reaction chamber. Water is heated by spraying it into the upper steam space. The proportionate amount of lime and soda is fed to the heating and mixing zone. Chemical reactions already described take place almost instantly. The precipitates are collected in collecting cone at the bottom and discharges to the sewer periodically. The

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softened water leaves the settling tank and then is further fed to the anthracite filters for further polishing.

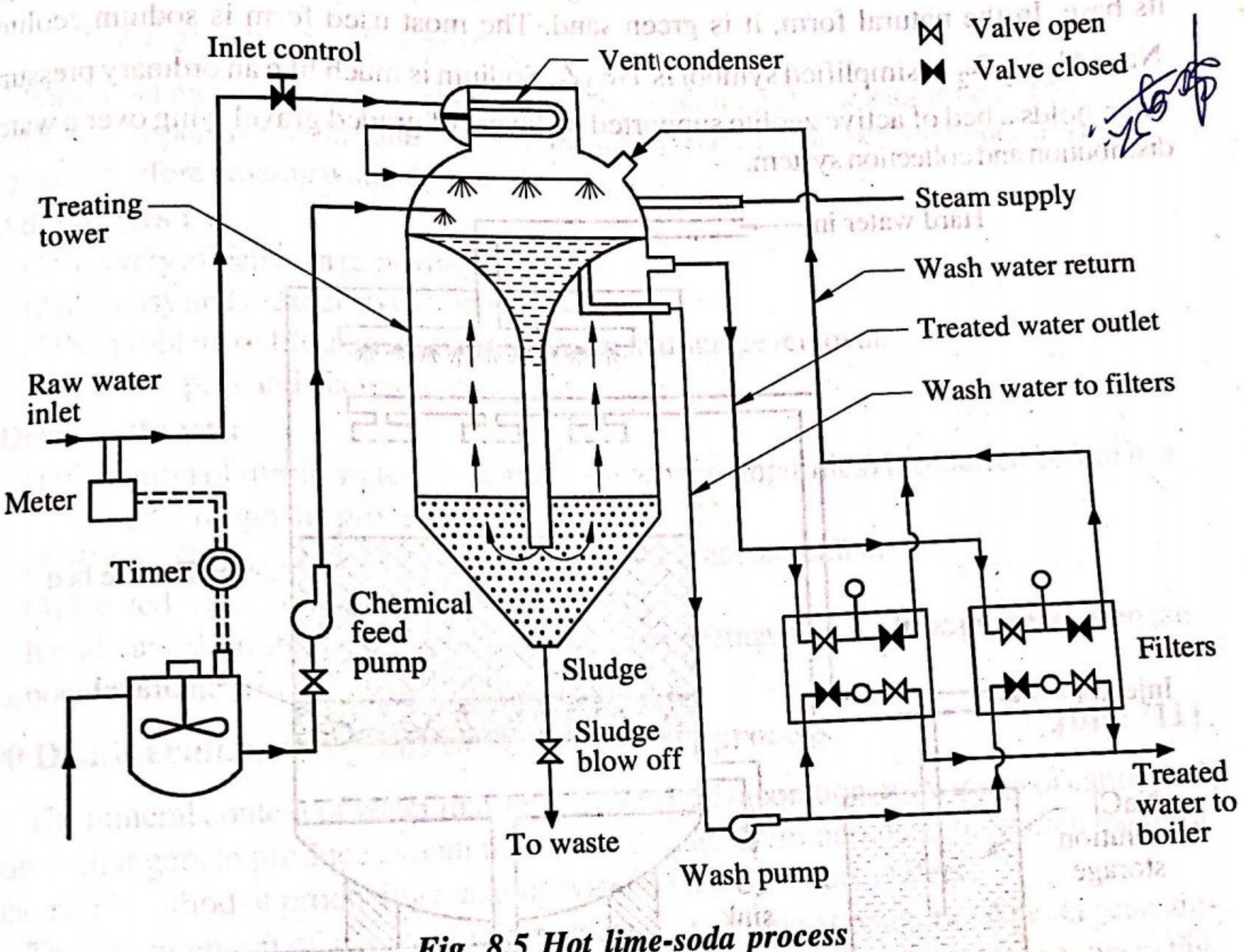


Fig. 8.5 Hot lime-soda process

- **Advantages :**
  (1) The silica is effectively removed from the water.
  (2) It is capable of treating large quantities of water in a relatively small unit.
  (3) It is more economical.
  (4) The size of the settling tank is reduced as the precipitate is larger in size and settles more quickly.
- **Disadvantages :**
  (1) Use of this method is limited to only low hardness water.
  (2) The softened water must be filtered before use to avoid to carry the precipitate with water.
  (3) Controlling of hot process is difficult.
  (4) The removal of the precipitate formed is difficult.

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8.9 Zeolite treatment [Jan. '13, Dec. '11]

Zeolite is either a natural or an artificial material that has property of being able to exchange its base. In the natural form, it is green sand. The most used form is sodium zeolite, $Na_2(Al_2Si_2O_2)$, simplified symbol is Na_2Z. Sodium is much like an ordinary pressure filter, holds a bed of active zeolite supported by layers of graded gravel lying over a water distribution and collection system.

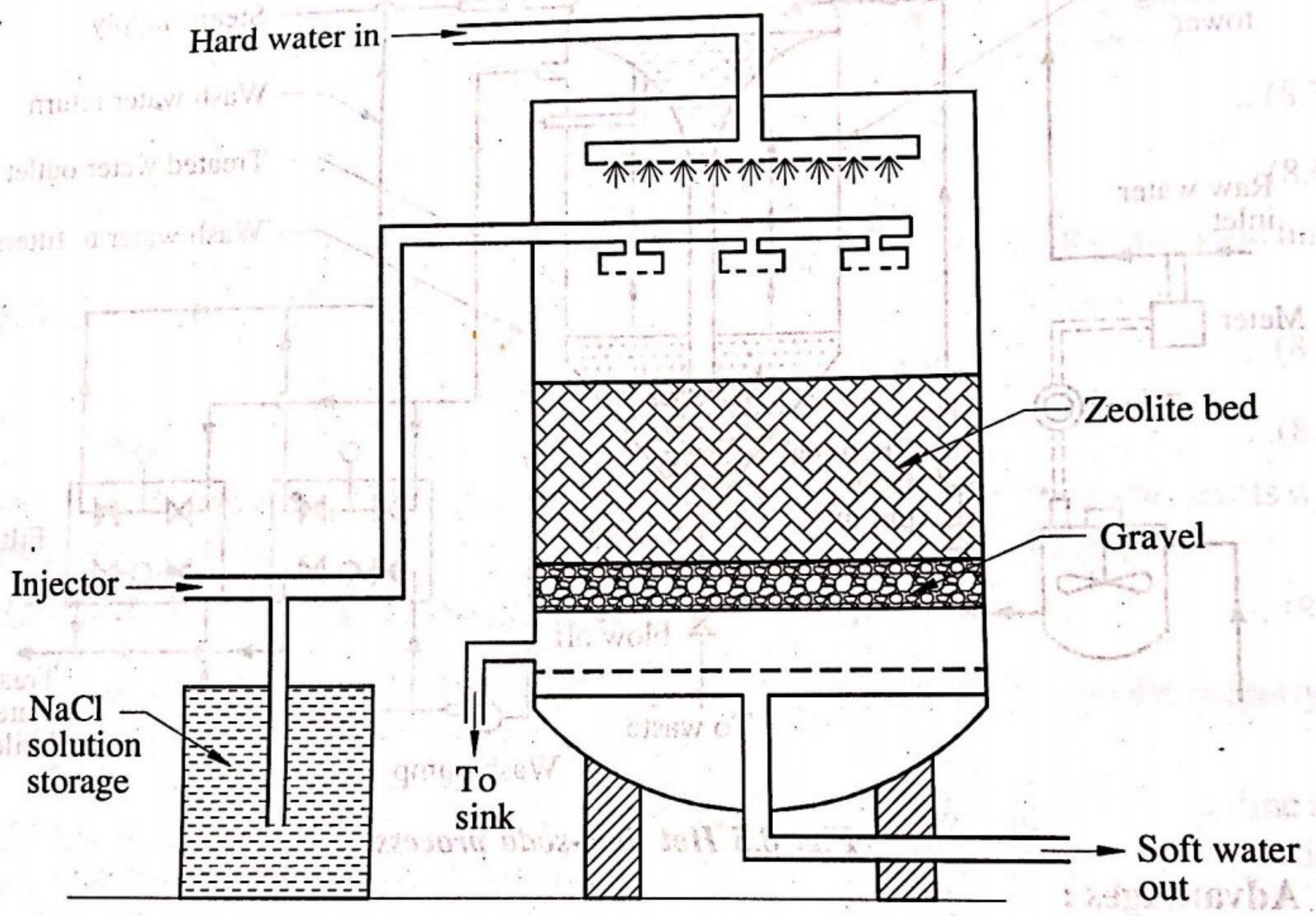


Fig. 8.6 Zeolite softner

The zeolite softener is shown in Fig. 8.6. The water sprayed at the top of the shell flows downward through the zeolite bed and the hardness of the water is removed by ion exchange. The chemical reactions can be written as :

$$Ca(HCO_3)_2 + Na_2Z = CaZ + 2NaHCO_3$$

$$CaSO_4 + Na_2Z = CaZ + Na_2SO_4$$

$$CaCl_2 + Na_2Z = CaZ + 2NaCl$$

Similarly, the hardness of carbonates and sulphates of magnesium, iron, manganese can be removed by sodium zeolite treatment.

The hard water flows through the zeolite bed and emerges as soft water. The zeolite is reactivated by back-washing the bed with suitable brine (NaCl solution). The regeneration

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process is given by the following equations :

$$CaZ + 2NaCl = Na_2Z + CaCl_2$$

$$MgZ + 2NaCl = Na_2Z + MgCl_2$$

Water containing suspended impurities can not be admitted to the zeolite bed, otherwise it will block the pores of zeolite and make zeolite inactive. Hence, suspended impurities must be removed before passing water through it.

- **Advantages :**
  (1)It is very effective in removing hardness.
  (2)It is easy and inexpensive in operation.
  (3)No problem of filtration of softened water and sludge removal.
  (4)It is compact and occupies less space.
- **Disadvantages :**
  (1)Filtration of turbid water (containing suspended impurities) is required before it is passed through the process.
  (2)Water of high or low pH value has deterious effect on zeolite.
  (3)Treated water contains more sodium salts.

It is always desirable to use zeolite process in combination with lime process when the bicarbonate hardness of the water is high.

8.10 Demineralization plant or Ion exchange process [June '11]

The mineral content of water may be removed by evaporation or by series of cation and anion exchangers to produce essentially distilled water. Demineralisation is often the most economical method of producing make-up water for high pressure boilers.

The arrangement of catexer (acid-regenerated cation exchanger) - Anexer (caustic-soda-regenerated anion absorber), with degasifying tower and silica absorber is shown in Fig. 8.7.

In this system, raw water is first passed through a weak acid cation exchanger to remove the bicarbonates. Water coming out contains dilute carbonic acid, hydrochloric acid and sulphuric acid. The water coming out of catexer is passed through Anexer. In Anexer, anions such as chlorides, sulphates and nitrates are removed from water with certain resinous materials (RCO_3). Anexer also exchanges acid radicals as given by the following equation :

$$2HCl + RCO_3 = RCl_2 + H_2CO_3$$

The water coming out of Catexer and Anexer is practically free from dissolved solids. The reactivation of anexer is done by circulating the solution of NaOH or Na_2CO_3 through the Anexer bed. The chemical reaction is given below :

$$RCl_2 + Na_2CO_3 = RCO_3 + 2NaCl$$

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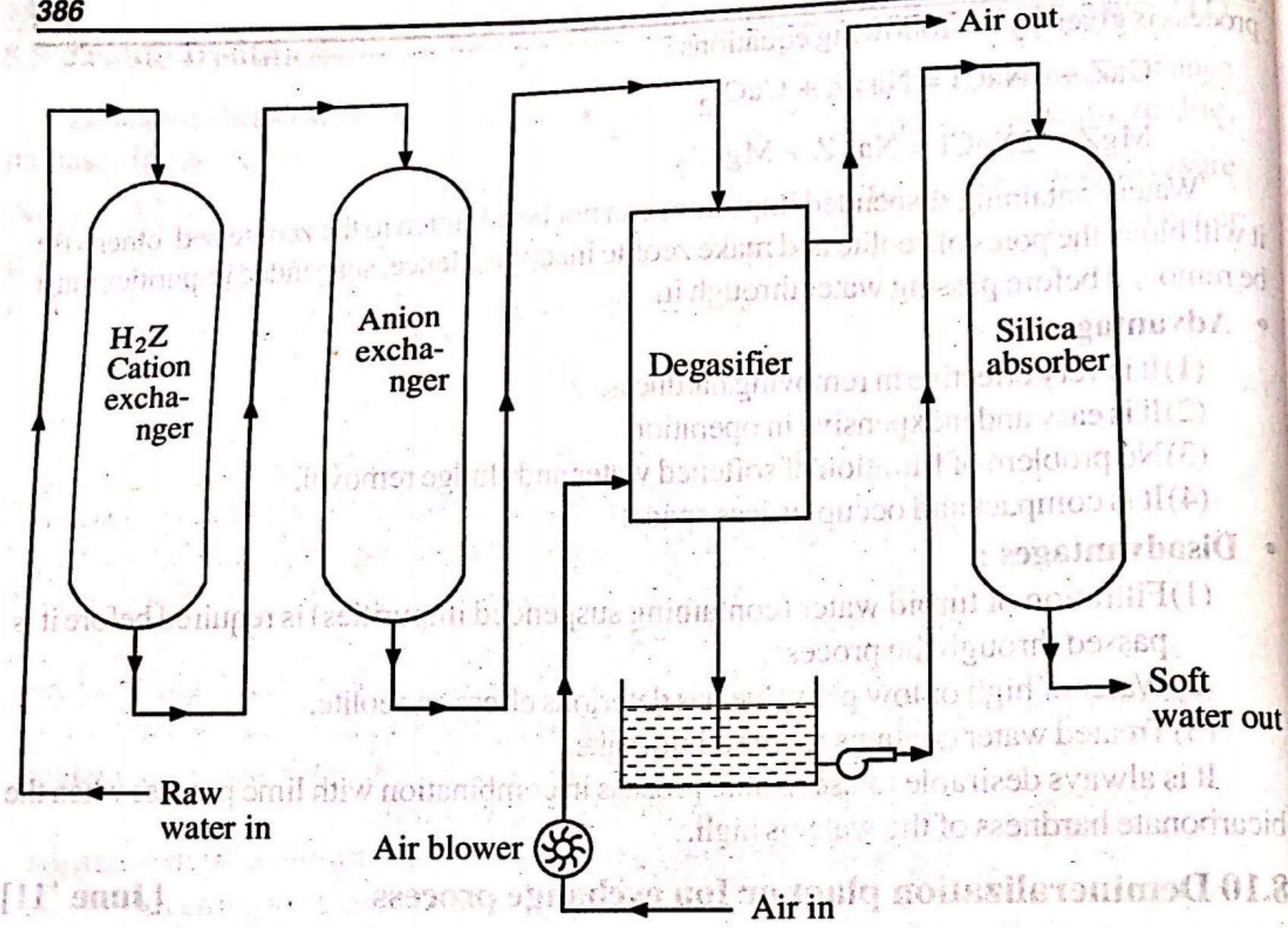


Fig. 8.7 Demineralizing process

The water coming out from Anexer is passed through a degasifier tower. This degasifier tower is filled with porcelain packing. Water is distributed over very large surface area provided by the packing as it flows downward through the tower. As the water comes in contact with low pressure air blown in at the bottom, CO_2 is liberated and is vented from the top of the tower. The degassed water containing about 2 to 5 ppm CO_2 is collected in the sump as shown in Fig. 8.7. The water is then passed through strong base-anion resin tower. This resin reduces the silica to 0.02 ppm. The reaction is given below :

$$R_4NOH \text{ (strong anion exchange material)} + H_2SiO_3 = R_4NHSiO_3 + H_2O$$

Strong base anion exchangers are specially designed to remove dissolved silica.

- **Advantages :**
  (1) It reduces boiler blow down and helps for the production of higher quality steam.
  (2) It supplies the purest quality feed water irrespective of flow quantity and its quality.
  (3) It can supply high quality feed water for the period needed to bring about an orderly shutdown to repair the condenser.

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- **Disadvantages :**
  (1) It's equipment cost is high.
  (2) It's operating cost is high.

8.11 Reverse Osmosis process

[May '13]

Reverse Osmosis (RO) is a pressure driven membrane process capable of removing small particles, dissolved salts and low molecular weight (100 mol. wt. cut-off) organics from a feedwater stream.

Principle :

If two aques solutions containing differential salt contents are kept separated by a semipermeable membrane, the system develops an inherent tendency for water molecules(not the salt) to permit across the membrane layer to dilute the more concentrated solution. This process is called ***osmosis***. The driving force per unit membrane cross-section, dependent on the concentration difference, is the ***osmotic pressure***.

However, this natural process can be reversed if some external pressure higher than the system's osmotic pressure is applied on the concentrated solution side whereupon water will pass from the more concentrated solution, through the membrane boundary, to the less concentrated solution. This is the working principle of reverse osmosis.

If one compartment contains pure water and the other salt or mineral-rich water, the process of reverse flow will produce pure water, since the membrane is not permeable to salt. This principle is shown in Fig. 8.8.

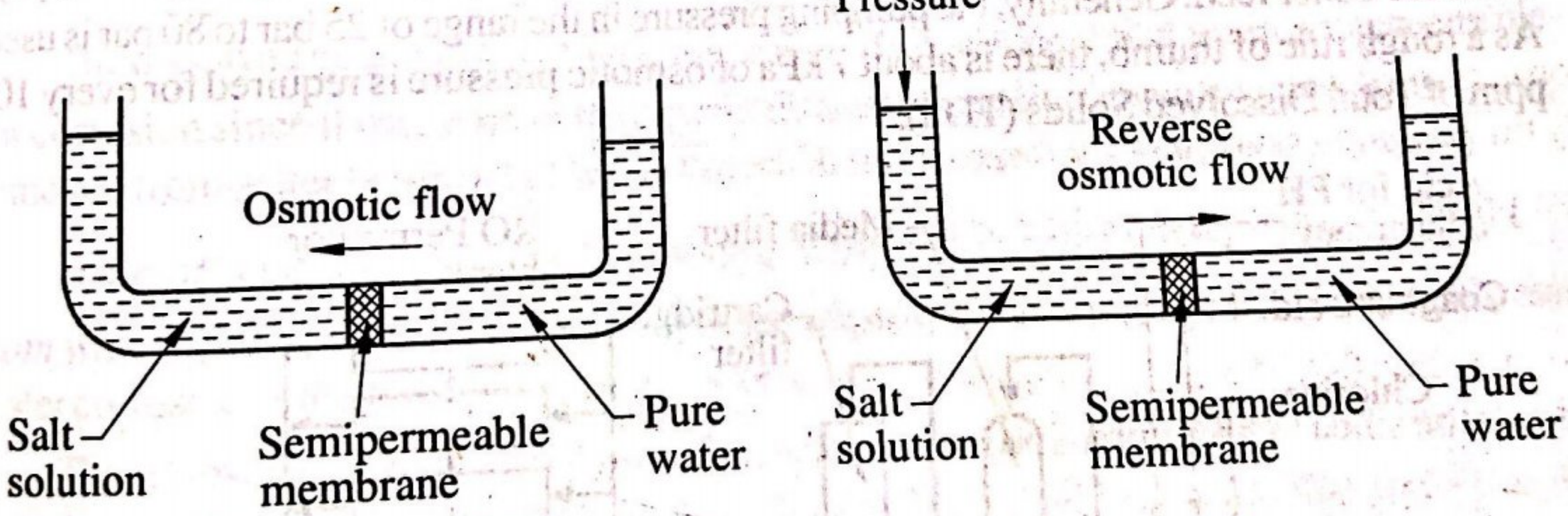


Fig. 8.8 Principle of Reverse osmosis

Working :

In actual practice, RO devices employ crossflow filtration as shown in Fig. 8.9. This method uses a pressurized feed stream which flows parallel to the membrane surface. Only a portion of this feed stream passes through the membrane as permit and the remainder of the stream known as the reject or concentrate provides a continuous sweep of rejected ions and particles from the membrane surface.

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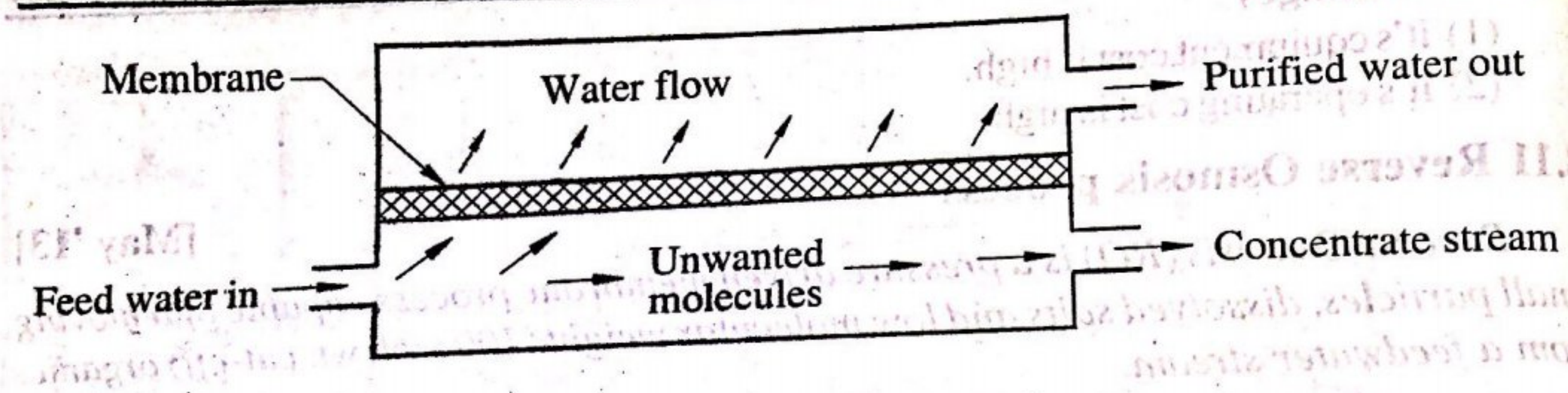


Fig. 8.9 RO cross flow filtration

8.12 Sea water treatment using reverse osmosis [June '11, Dec. '11]

The arrangement of sea water treatment using reverse osmosis is shown in Fig. 8.10. Depending upon the clarity of the raw water, pretreatment of sea water is required before passing through RO systems. First chlorine is passed through the water to oxidise organic products and kill marine life. The aluminium sulphate (coagulant) is added to water inorder to precipitate colloidal material and colour. Then H_2SO_4 or HCl is added to reduce the pH value to about 6 and suppress calcium carbonate scaling. Then water is passed through sand filters and activated carbon filters, where it is filtered. The chemical treatment is further carried out to suppress carbonate sealing by adding sodium hexameta phosphate ($NaHSO_3$). Finally water is passed through 5 μm cartridge filter to remove remaining suspended matter. The water is then pumped through RO system with the help of high pressure pump to get pure water for boiler feed. Generally, the pumping pressure in the range of 25 bar to 80 bar is used. As a rough rule of thumb, there is about 7 kPa of osmotic pressure is required for every 100 ppm of Total Dissolved Solids (TDS).

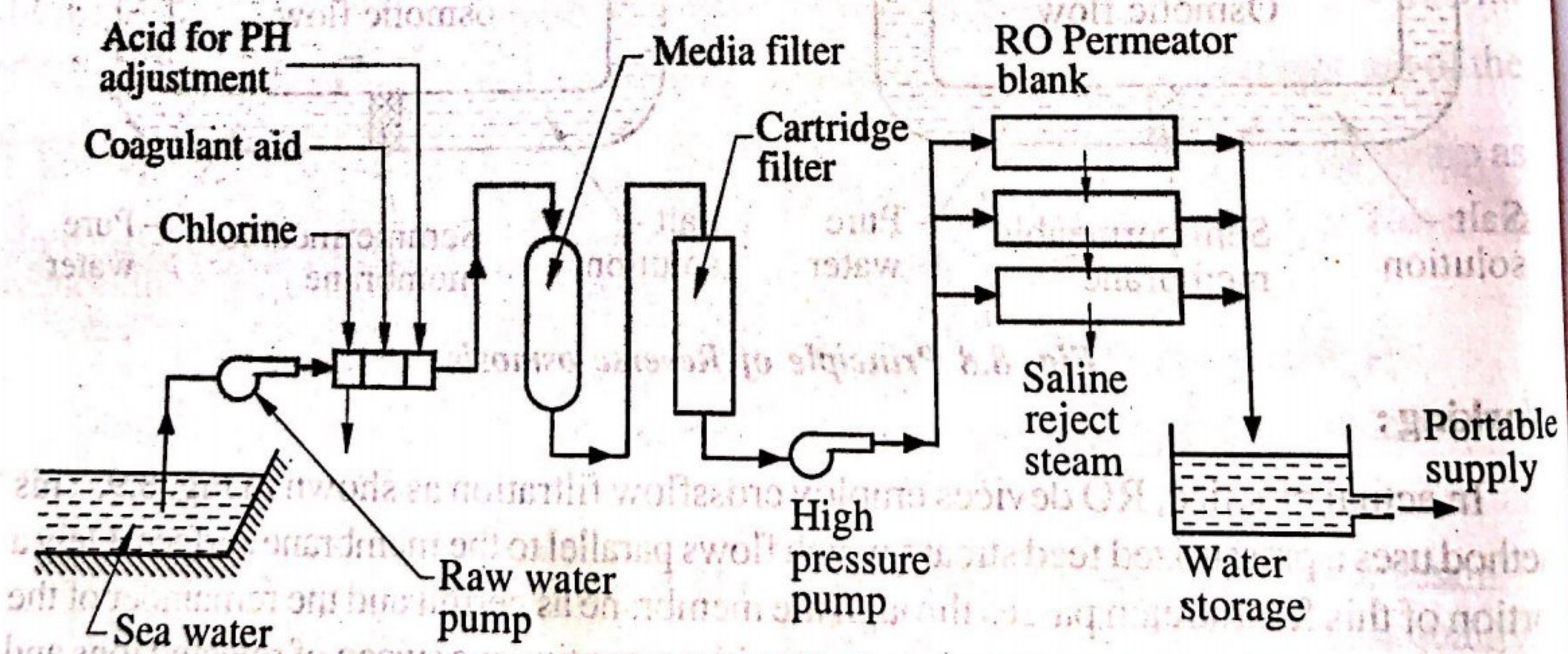


Fig. 8.10 Sea water treatment using RO

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Membrane of RO system consists of a polymeric material film made of proper porosity, from materials like acrylics, polyamides, aramids, etc. Special attention is needed to select the materials for construction of various parts as sea water is highly corrosive. Glass-reinforced plastic, cupro-nickel, monel and nickel alloys are generally used. In addition to this, care must be taken in siting the sea water intake to avoid ingress of silt and weed.

This method can be employed more economically in the range of dissolved solids from 2,000 ppm to 10,000 ppm or more. This process may be more competitive with thermal distillation for the conversion of sea water into boiler make up feed.

- **Advantages of reverse osmosis system :**

(1) It is the most efficient means for removing salts, chemical contaminants and heavy metals such as lead from water.
(2) It is less expensive than demineralization for waters with total dissolved solids of 200 ppm or more.
(3) It is simple to operate and maintain.

- **Disadvantages :**

(1) It requires periodic membrane cleaning.
(2) It's membranes need to be changed every one to three years depending on water quality, size of the system and pretreatment.
(3) It's initial cost is high.

8.13 De-aeration

The dissolved gasses in water like O_2, CO_2, H_2S, air, and other gases are responsible for corrosion since these react with impurities and form acids. Therefore, these gases are removed from water before supplied to the boiler by the method of ***thermal treatment***.

Deaeration is the process of removing dissolved corrosive gases (O_2 and CO_2) from water. This process is also called ***degasification***. These corrosive gases make the water corrosive as they react with metal to form iron oxide.

The removal of gases is accomplished by heating the water to 105-110°C with subsequent agitation during heating. The simple heating removes the dissolved gases from water because the absorbing capacity of water is reduced at higher temperatures.

The arrangement of tray type deaerator is shown in Fig. 8.11. It consists of a deaerator column fitted on the top of the storage tank. The deaerator column is hollow cylindrical vessel provided with horizontal perforated trays, water distribution device at the top, and steam distributor at the bottom.

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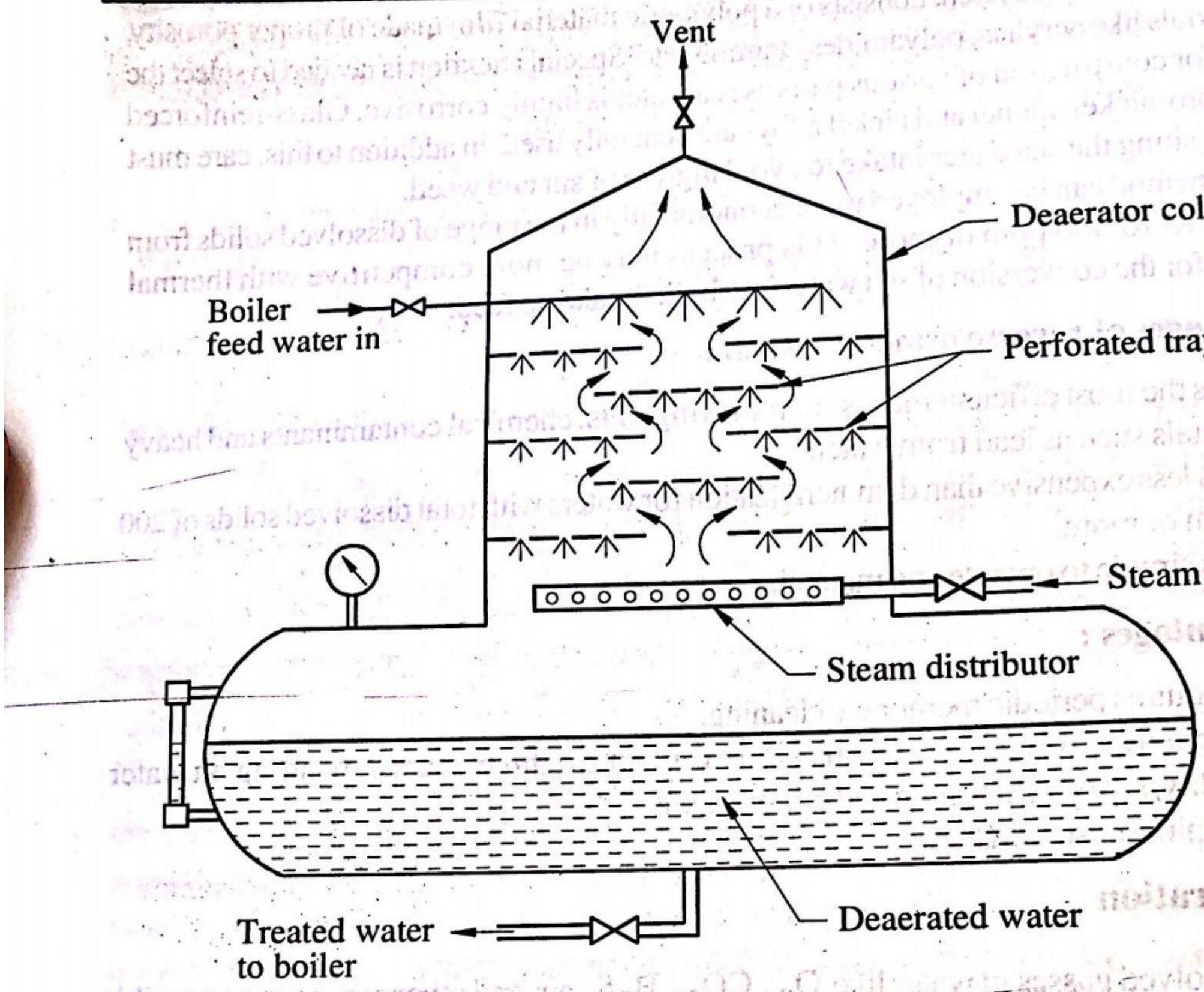


Fig. 8.11 Tray type deaerator

Feed water enters the deaerator from the top and is evenly distributed by means of the distributing device into the perforated trays fitted at the bottom. Water fills the perforations (dia. 5-7 mm) and rains down and comes in contact with heating steam delivered into the lower portion of the deaerator column through the steam distributor. As a result of heat exchange between the steam going up and the feed water stream flowing down, the water gets heated up to its boiling point and the gases (O_2, CO_2) dissolved in it are transferred to the gas phase. These gases together with non-condensing vapour are vented into the atmosphere through a vent valve or pass into a vent condenser where the steam is condensed and the O_2 and CO_2 are vented.

The deaerated water is collected in the storage tank which is fitted with gauge glass and pressure gauge to avoid the formation of high pressure or vacuum in the deaerator.

With this type of deaerator, the oxygen content can be reduced below 0.005 CC per litre. As CO_2 is also removed, the increase in pH value of water also give an indication of deaeration efficiency.

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Exercise

8.1 What are the basic objectives of feed water treatment ?
8.2 What are the impurities in raw water ?
8.3 Explain the necessity of "Feed water treatment". What are the major impurities present in feed water ?
8.4 Explain the effect of impurities in boiler feed water.
8.5 Explain the terms "Priming and Foaming" in feed water ? What are its effects on boiler? Explain the effects of dissolved gases on boiler.
8.6 Explain the mechanism of scale formation in boiler.
8.7 What do you understand by caustic embrittlement ?
8.8 What is pH value ? How is it controlled in water ?
8.9 Explain the term 'pH' value. Why boiler feed water should be preferred with $9 > pH > 10$?
8.10 What is the role of pH value on scale formation ?
8.11 What is internal treatment of feed water ? How is it done ?
8.12 Why is trisodium phosphate injected into the boiler drum ?
8.13 What is blow down ? Why is it needed ?
8.14 What do you understand by external treatment of feed water ?
8.15 Write short notes on Hot soda lime treatment for feed water.
8.16 Explain sodium zeolite feed water treatment in thermal power plants with diagram.
8.17 Explain Demineralization plant with neat sketch.
8.18 State advantages and disadvantages of demineralization plant.
8.19 Explain the principle and working of a reverse osmosis plant for feed water treatment.
8.20 Explain the term pH value of water and its importance in feed water. Explain sodium zeolite water treatment method with diagram.
8.21 Explain sea water treatment using reverse osmosis process.
8.22 Explain the role played by "oxygen" in corrosion of components in power plant equipment. How "oxygen" can be scavenged ?
8.23 Write short note on deaerator.

Objective Type Questions

1. In boilers, the feed water treatment is done mainly for removing _____ troubles.
(a) corrosion (b) scale formation (c) embrittlement (d) all of the above

2. The scale formation in boiler
(a) reduces heat transfer rate (b) raises the temperature of the metal wall
(c) reduces flow area (d) all of the above.

3. _____ is the passing of small water particles with steam as it leaves the boiler.
(a) Foaming (b) Priming (c) Carry over (d) All of the above

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4. _____ is the passing of dissolved solids with steam while it leaves the boiler.
(a) Foaming (b) Priming (c) Carry over (d) All of the above

5. _____ is caused by using highly alkaline water.
(a) corrosion (b) embrittlement (d) carryover (d) scale formation

6. Embrittlement is not caused by following condition:
(a) when the boiler water contains dissolved oxygen and CO_2.
(b) when the boiler water contains free hydroxide alkalinity.
(c) slow leakage of boiler water through a joint.
(d) boiler metal is highly stressed at the point of leakage.

7. The pH value of feed water should be maintained greater than _____ to reduce corrosion.
(a) 9.6 (b) 10.6 (c) 7 (d) 6.7

8. To avoid scale formation _____ is added to boiler drum when alkalinity of boiler water is low.
(a) disodium phosphate (b) mono-sodium phosphate
(c) trisodium phosphate (d) sodium carbonate

9. Blowing down of boiler water is the process to
(a) reduce the boiler pressure
(b) reduce the boiler temperature
(c) increase the boiler pressure
(d) control the solid concentration in the boiler water by removing some of the concentrated saline water.

10. The following is not true for hot lime soda process
(a) calcium hydroxide and sodium carbonate are used as treating elements.
(b) It is suitable for higher hardness water.
(c) the softened water must be filtered before use.
(d) It removes silica very effectively.

11. Deaerative heating is done to
(a) preheat the water (b) preheat the air
(c) remove dissolved gases in the water (d) heat the air in the water

Answers

1. (d) **2.** (d) **3.** (b) **4.** (c) **5.** (b) **6.** (a) **7.** (a) **8.** (c) **9.** (d) **10.** (b)
11. (c)

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GTU Study Materials ☑

Chapter 9
Gas Turbines

9.1 Introduction
9.2 Basic working principal of gas turbine
9.3 Advantages and disadvantages of gas turbine
9.4 Applications of gas turbines
9.5 Classification of gas turbines
9.6 Simple open cycle gas turbines
9.7 Closed cycle gas turbine
9.8 Comparison between the open cycle and closed cycle gas turbine
9.9 Gas turbine fuels
9.10 Ideal Brayton cycle
9.11 Work ratio and air rate
9.12 Actual Brayton cycle
9.13 Optimum pressure ratio for maximum work output
9.14 Optimum pressure ratio for maximum thermal efficiency
9.15 Effect of operating variables on thermal efficiency
9.16 Effect of operating variables on air rate
9.17 Effect of operating variables on work ratio
9.18 Means of improving efficiency and specific work output of simple open cycle gas turbine
9.19 Effect of various modifications
9.20 Water injection
9.21 Combustion chambers
9.22 Combined steam and gas turbine power plant
9.23 Different arrangement of combined cycle power plant
9.24 Thermodynamic analysis of combined cycle
9.25 Gas Turbine blade cooling

9.1 Introduction

A gas turbine is the most satisfactory way of producing large power in compact unit. In gas turbine power plant, a gas turbine is used as the prime mover to produce mechanical energy. In the gas turbine power plant, the working medium is either a mixture of combustion products (air and fuel) or heated air at a certain pressure and a higher initial temperature. The gas turbine obtains its power by utilizing the energy of burnt gases and air which are at high temperature and pressure by expanding through the several rings of fixed and moving blades.

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The simple gas turbine plant consists of (i) a compressor, (ii) a combustion chamber and (iii) a gas turbine.

The basic principle on which a gas turbine works in similar to internal combustion engine. In both the cases air is made to enter the prime mover which compresses and then heated by combustion process. Hence pressure and temperature of working fluid is increased. This high pressure and temperature working fluid is then expanded in the prime mover. The expanded products are discharged through the exhaust. However in the case of internal combustion engine is reciprocating machine in which charge induces into the engine cylinder intermittently. In case of gas turbine, flow of working fluid through the rotary machine (gas turbine) is continuous and smooth. With respect to design, a gas turbine is quite similar to a steam turbine. In the turbine blading working gas expands and the heat energy is converted first into the kinetic energy and then to the rotary motion of the turbine shaft. However, in steam power plant the products of combustion do not form the working medium. These are utilized to produced an intermediate fluid (steam) which is expanded in the steam turbine.

9.2 Basic working principle of gas turbine

In a gas turbine first air is obtained from the atmosphere and compressed in an air compressor as shown in Fig. 9.1. This high pressure air is then passed into the combustion

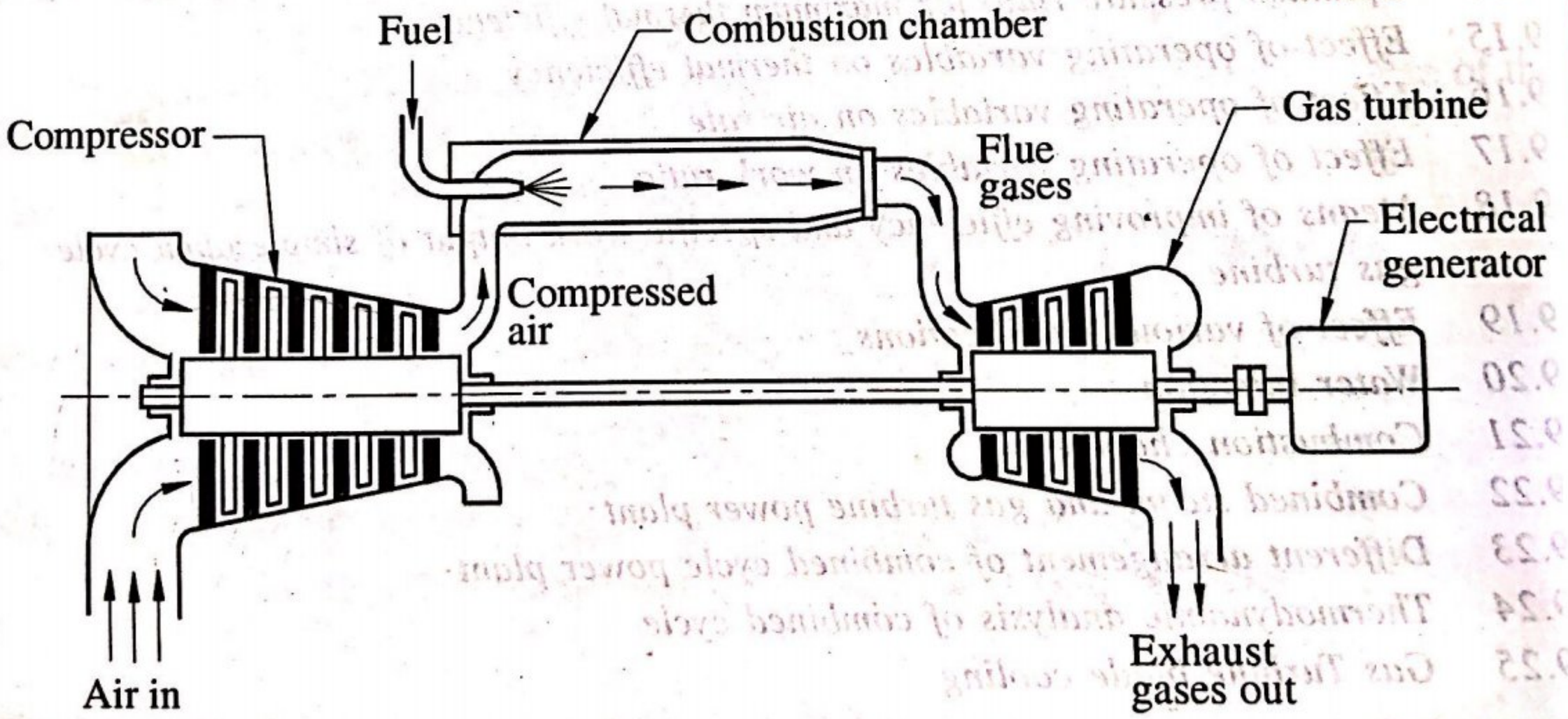


Fig. 9.1 Simple cycle gas turbine power plant

chamber, where it is heated due to combustion of fuel. The product of combustion (hot gases) of high pressure and temperature passes through the passages formed by the stationary and rotating blades of gas turbine. A jet of hot gases is made to flow over rings of blades imparting rotary motion to the shaft of the turbine. A larger part of the power developed by the turbine rotor is consumed for driving a compressor which supplies air under pressure to combustion chamber, while remaining power is utilized for doing the external work.

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9.3 Advantages and disadvantages of gas turbines

(1) Advantages and disadvantages of gas turbines over steam turbines

Advantages :

(i) It is simpler in design since there are no boilers, no condensers, feed water supply, no cooling tower. Hence weight and space of gas turbine are less than those of a steam turbine. A gas turbine is more compact than a steam turbine of the same output and hence lower capital costs.

(ii) The handling of ash is a major problem in steam turbine plants which complete eliminated in open cycle gas turbine plants using gas or liquid as fuel.

(iii) It can be useful in the region where large amount of water is not available for steam power plants.

(iv) Gas turbine power plant can be installed near to load centre since open cycle gas turbine, no cooling water is required and cubic capacity of the buildings required for gas turbine plant is about 50% of steam plants.

(v) The delivery and installation time for gas turbine power plant is much less than that for steam power plants.

(vi) Suitable blade cooling permits the use of high gas turbine inlet temperature (as high as 1300°C) yielding a high thermal efficiency (about 37%).

(vii) The gas turbine exhaust can be used to produce process heat for various uses.

(viii) A gas turbine plant can be used in conjuction with a bottoming steam plant in the combined cycle mode to yield on overall fuel to electricity efficiency of 55%.

(ix) The gas turbine plant can be started quickly and can be put to share full load within a few minutes.

(x) Gas turbines may be operated by remote control and need little or no personnel while operating and while shutdown.

(xi) The gas turbine plant uses fewer auxiliaries than the steam turbine plant, therefore, small size of the gas turbine components enables complete work tested units to be transported to the site.

Disadvantages :

(i) The gas turbine unit has got starting problems as the compressor has to be started before starting of turbine and the former requires power from some external source.

(ii) The temperature of combustion chamber is quite high and therefore its life is comparatively reduces.

(iii) High air rate is required to limit the maximum gas turbine inlet temperature as a result of which the exhaust losses are high unless the waste heat in it is utilized.

(iv) Larger part of the turbine work available from turbine is used up in driving the compressor. Hence the net work available is less.

(v) Part load efficiency is low. Also, plant efficiency is highly sensitive to component

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efficiencies like turbine efficiency, compressor efficiency, combustion chamber efficiency, etc.

(2) Advantages and disadvantages of gas turbines over internal combustion engine

Advantages:

(i) The gas turbine being a rotary machine. Due to absence of reciprocating parts, the vibrations are absent resulting in better and easy balancing and small foundations.

(ii) Due to less number of parts, is easier to maintain. The maintenance costs of gas turbine is less (about 60 %) compared to diesel engine power plant.

(iii) High specific power output and weight of gas turbine plant is much less compared to I.C. engine plant. The weight power ratio of gas turbine plant is about 0.18 kg/hp while it is 0.5 kg/hp for I.C. engine.

(iv) The cooling requirement of gas turbine is very less compared to that of I.C. engine. . absence of radiator is a big advantage in case of automotive use of gas turbine.

(v) The capacity of I.C. engine is limited due to detonation or knocking and low volumetric efficiency at high speed. The capacity of gas turbine can be increased with use of highly developed materials (i. e. high inlet temperature) of gas turbine).

(vi) The gas turbine requires less lubricating oil as compared to diesel engine power plant. The cost of lubricating oil in case of gas turbine is about 0.5 to 1 % of the total fuel cost while it is about 10 to 15 % that of for diesel engine.

(vii) The closed cycle gas turbine makes possible to use any fuel ranging from kerosene to heavy oil and even peat and coal slurry.

(viii) A gas turbine plant can be used in conjuction with a bottoming steam plant in the combined cycle mode to yield on overall fuel to electricity efficiency of 55%.

(ix) The mechanical efficiency of gas turbine is more than that of for I.C. engine.

Disadvantages:

(i) The thermal efficiency of simple gas turbine plant is lower than that of I.C. engine. In order to get efficiency that of I.C. engine gas turbine should be provided with additional auxiliaries such as intercooler, regenerator and reheater. Resulting in complex system and higher costs of plant.

(ii) The gas turbine is high speed engine and hence it requires reduction to reduce the rorational speed for industrial applications

(iii) The temperature of combustion chamber is quite high and therefore its life is comparatively reduces. Also, the cost of manufacture of a gas turbine plant is higher due to use of high heat resistant materials and special manufacturing process used for blades.

(iv) High air rate is required to limit the maximum gas turbine inlet temperature as a result of which the exhaust losses are high unless the waste heat in it is utilized.

(v) Part load efficiency of gas turbine is poor. Also, plant efficiency is highly sensitive to component efficiencies like turbine efficiency, compressor efficiency, combustion chamber efficiency, etc.

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(vi) The gas turbine plant is not self starting unit. Also, it has starting problems as the compressor has to be started before starting of turbine and the compressor requires power from some external source.

9.4 Application of gas turbines

The continued developments in gas turbine materials toward increasing the turbine inlet temperature and improving the components efficiencies over the past decade has enlarged the spectrum of gas turbine applications to area such as power generation, utility industry, space, marine and automotive propulsion etc.. The main reasons of such a wide popularity being employed by the gas turbine is its advantages of simplicity, high power/weight ratio, smooth running, less maintenance, multi-fuel capability, suitability for combined cycle and reliability.

The gas turbines applications can be divided in the following categories :

(1) For electric power generation : The gas turbine are very popular for electric power generation because of the ability of starting and brought upto full load quickly and less cost of installation and maintenance.

- Majority of the gas turbine power plants are used for **peak load** service with other types of power plants (steam and hydro power plants)
- A large number of gas turbine plants is also used as **stand by** power plants in hydro-electric power plants.
- Gas turbine power plants are also used as **base load** power plants where fuel oil or natural gas are cheap and easily available, water supply is scarce and load factor is very low (15-18%).
- Now a day gas turbine plants are used to operate as combination plants with steam plants, called as **combined cycle power plant.**

(2) For jet propulsion engine : Every turbojet and turbo-propeller engine has a gas turbine. The turbine supplies power only to drive the air compressor in the turbojet engines while in turbo-propeller engines they may drive the propeller in addition to the compressor.

(3) For supercharging : A small gas turbine run by the hot exhaust gases which is used to run supercharger (compressor) for aviation gasoline engines and for heavy duty diesel engines.

(4) For marine field : A gas turbine can be also used for propulsion of ships or power generation on the ships.

(5) Railway and road transport : Gas turbine can be also used for rail propulsion and high speeds road vehicles like racing cars.

(6) Industry : Gas turbines are also employed for industrial applications like blast of air for blast furnace in steel industries, oil and other chemical industries. Gas turbine is also used to supply preheated combustion air, hot gases for an industrial process in petrochemical industries.

9.5 Classification of gas turbines [Nov. '14]

The gas turbine can be classified as follows :

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(1) According to types of combustion process

(i) Constant volume or explosion type gas turbine : This type of gas turbine works on Atkinson cycle in which combustion of air fuel mixture is takes place at constant volume. In the constant volume combustion process, the fixed volume of air and fuel burns at constant volume, hence air and fuel mixture should be isolated from co npressor. This is possible by valves in the combustion chamber, resulting in an intermittent combustion which inherently impairs smooth running of machine. This types of turbine has better thermal efficiency than a constant pressure cycle gas turbine. The main disadvantages of this type of gas turbine is that complexity in mechanical system and pressure difference and velocities of hot gases are not constant, so turbine speed fluctuates. The constant volume combustion type gas turbine is not popular in practical use.

The constant volume gas turbines are absolute in use. Therefore, in this chapter we will discuss only constant pressure gas turbines.

(ii) Constant pressure or continuous combustion type gas turbine : This type of gas turbine works on Brayton cycle in which combustion of air fuel mixture is takes place at constant pressure.

(2) According to direction of flow

(i) Axial flow gas turbine

(ii) Radial flow gas turbine

(3) According to action of expanding gases

(i) Impulse gas turbine

(ii) Impulse- Reaction gas turbine

(4) According to path of working substance

(i) Open cycle gas turbine plants : In the open cycle gas turbine, ambient air enters at the compressor and after the compression of air, fuel is burned in the air itself to raise it to a high temperature and then product of combustion are passed on to the turbine for expansion and which after delivering the work are finally rejected to atmosphere. In the open cycle the working medium is continuously replaced by fresh air and fuel.

(ii) Closed cycle gas turbine plants : In the closed cycle gas turbine power plant, the same air or the working fluid is circulated over and over again. The working medium is not mixed with the fuel, but it is heated by the burning of fuel in a separate supply of air in the combustion chamber and transferring this heat to the working fluid which passes through tubes fitted in the combustion chamber. The working fluid does not come into direct contact with products of combustion.

(iii) Semi-closed cycle gas turbine plants : Semi-closed cycle gas turbine plant is combination of open and closed cycles, in which some part of working fluid is recirculated to the plant and another part of working fluid flows into and from the atmospheric air.

(5) According to thermodynamic cycle

(i) Simple cycle

(ii) Simple cycle with intercooling : In which the air is cooled between stages of

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compression.

(iii) Simple cycle with regeneration : In which the air after compression is heated with help of exhaust gases coming from turbine.

(iv) Simple cycle with reheating : In which combustion products (gases) after part of expansion in high pressure turbine is reheated in second combustion chamber and then it is expanded in the low pressure turbine.

(v) Simple cycle with intercooling, regeneration and reheating

(6) According to shaft configuration

(i) Single shaft engine : - Examples: Turbojet engine, Turboshaft engine

(ii) Multi-shaft engine : - Examples: Two shaft engine with power turbine, Turbojet engine, Industrial slit shaft engine, Twin-spool engine

(7) According to applications

(i) Power or industrial gas turbines

(ii) Aviation or aircraft turbines

9.6 Simple open cycle gas turbine [Nov. '14]

A simple open cycle gas turbine consists of a (i) compressor, (ii) combustion chamber and (iii) gas turbine.

In the open cycle gas turbine, ambient air enters at the compressor and after the compression of air, fuel is burned in the air itself to raise it to a high temperature and then product of combustion are passed on to the turbine for expansion. After delivering the work combustion products are finally rejected to atmosphere. In the open cycle the working medium is continuously replaced by fresh air and fuel. It works on the Joule cycle or Brayton cycle.

A schematic arrangement for simple open gas turbine plant is shown in Fig. 9.2. The air is sucked in by the compressor from the atmosphere through the filter which removes the dust from the air. The rotary blades of the compressor push the air between the stationary blades to raise its pressure to 4-5 atmosphere. Hence air is available at high pressure at output of the compressor. Then high pressure air passes through combustion chamber, in which heat added to the air at constant pressure by burning the fuel and raises temperature (about 1650°C) of working medium. This high temperature must be brought down to a level so that the thermal stresses in the turbine blades do not become excessive. This is achieved by allowing the remainder air to enter the combustion chamber at downstream to mix and cool down the combustion gases. The products of combustion comprising of mixture of gases at high pressure and temperature are passed through the gas turbine. These gases in passing over the turbine blades expand and thus result in motion of rotor and finally discharged to the atmosphere at the temperature about 540°C.

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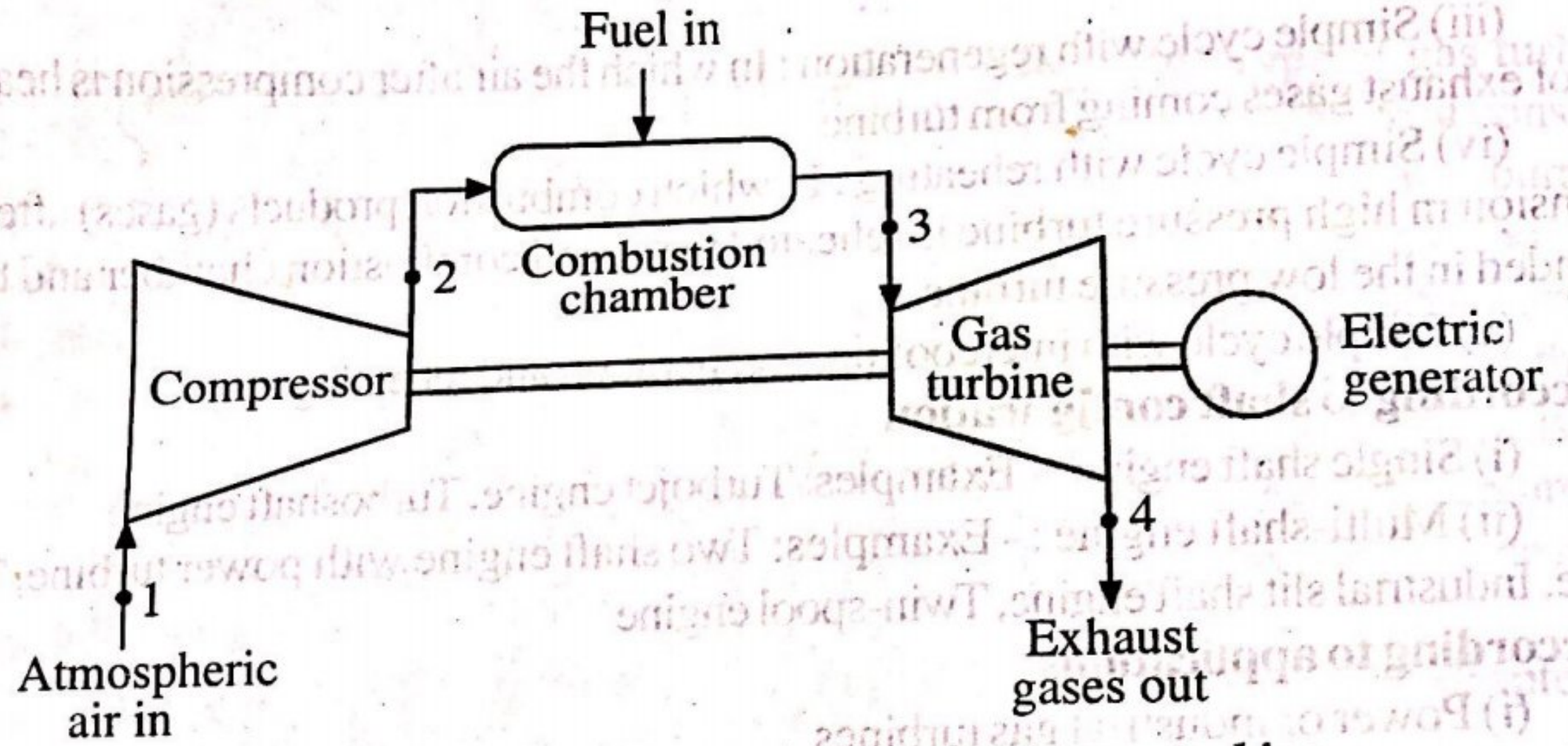


Fig. 9.2 Schematic diagram of open cycle gas turbine

Advantages of open cycle :

(i) Simplicity : There are only few rotating parts as turbine, compressor and gear train driving the auxilliaries. Hence problem of vibrations and lubrication is not so severe. The ignition system is also simple compared to closed cycle.

(ii) Flexibility : Since different processes within the cycle take part in separate components, a great variety in the arrangement of the system is possible.

(iii) Low weight and size : The weight in kg. per kW developed is less.

(iv) Independent system : Open cycle gas turbine power plant, except those with intercooler, does not require cooling water. Therefore the plant is independent of cooling medium and becomes self-contained.

(v) Fuels : Almost any hydrocarbon fuel from high octane gasoline to heavy diesel oils including some solid fuels can be burned in the combustion chamber.

(vi) Warm-up time : After the turbine has been brought up to speed by the starting motor and the fuel ignited the gas turbine will accelerate from cold start to full load without a warm up time. This is particularly important in stand by emergency plants.

Disadvantages of open cycle :

(i) Part load performance : The part load efficiency of the open cycle plant decreases rapidly as the considerable percentage of power developed by the turbine, is used to drive the compressor. Also the system is sensitive to the changes in component efficiency.

(ii) Sensitivity : Since system sensitive to the component efficiency, particularly that of compressor. The efficiency of compressor is affected by change in the atmospheric conditions such as temperature and humidity of air at the inlet and foreign matter contained in the air.

(iii) High air rate : The simple open cycle gas turbine has a very high air rate as compared

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to other prime movers. However, the air rate may be lowered by intercooling and reheating.

(iv) Erosion and corrosion : The working fluid is mixture of air and fuel. Since air contains dirt being deposited on the compressor blades. Due to carbon and other foreign deposits from combustion in the combustion chamber, turbine and regenerator, it is necessary that the dust should be prevented from entering into the compressor in order to minimised erosion and depositions on the blades and passages of the compressor and turbine.

(v) In the simple open cycle, the turbine exhaust is discharged into atmosphere. Since turbine exhaust contain large amount of heat resulting in loss of heat.

9.7 Closed cycle gas turbine [May '12]

As we know that in open cycle gas turbine power plant, the fuel is mixed with air in the combustion chamber and thè combustion gases are expanded in the gas turbine which causes erosion and corrosion of turbine blades and therefore it is necessary to use fuel superior quality in the combustion chamber in order to minimise erosion and corrosion. This negative effect is overcome in case closed gas turbine power plant. In the closed gas turbine power plant, the same air or the working fluid is circulated over and over again. The working medium is not mixed with the fuel, but it is heated by the burning of fuel in a separate supply of air in the combustion chamber and transferring this heat to the working fluid which passes through tubes fitted in the combustion chamber. The working fluid does not come into direct contact with products of combustion. The other disadvantages of open cycle plant is that the turbine exhaust is discharged into the atmosphere resulting in rejection of heat of exhaust gases to the atmosphere. In case of closed cycle these heat are recovered in a heat exchanger or recooler.

A schematic arrangement of closed cycle gas turbine plant is shown in Fig. 9.3. The working fluid (air or any other suitable gas such as helium, argon, hydrogen and neon) coming out from compressor is heated in a heat exchanger (heater) by an external source at constant pressure. The high temperature and high pressure air coming out from the external heater is passed through the gas turbine. The working fluid coming out from the turbine is cooled to its original temperature in the heat exchanger (cooler) using external cooling source before passing to the compressor. In the closed cycle, the working fluid is continuously circulated through compressors, cooler, heater and turbine without its change of phase, the required heat addition and rejection taken place in the heater and cooler respectively.

The performance characteristics, component elements and analysis for open cycle apply equally as well as to the closed cycle gas turbine.

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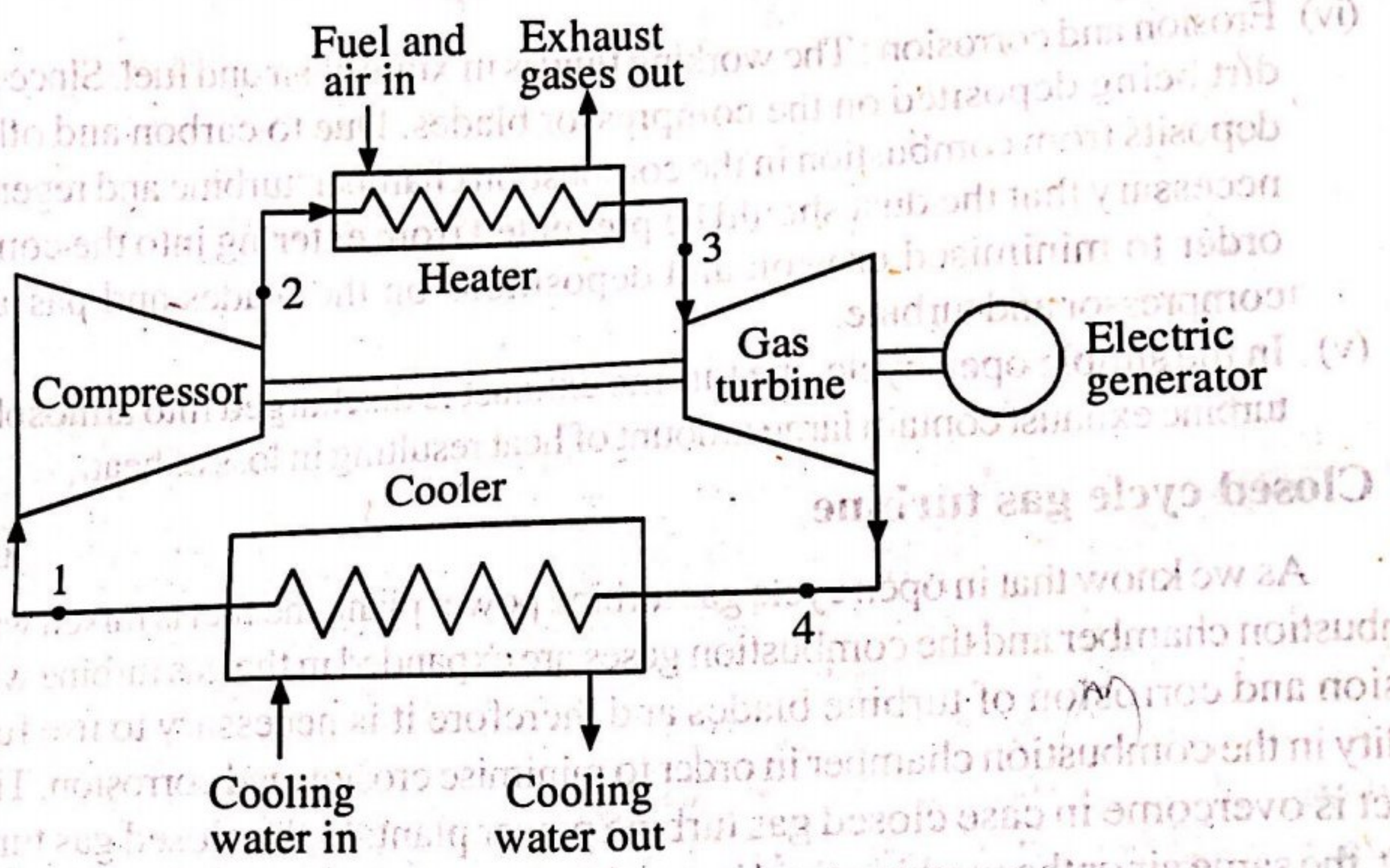


Fig. 9.3 Schematic diagram of closed cycle gas turbine

Advantages of closed cycle over open cycle : **[May '12]**

(1) In the closed cycle gas turbine the working medium is heated externally and the fuel does not mix with it, hence any inexpensive solid fuel such as coal can be used to heat the working fluid. Also need for filtration of the incoming air is completely eliminated.

(2) They are kept free from the products of combustion. Hence absence of corrosion and abrasion of the interiors of turbines and compressors extends the life of the plant and maintains the efficiency of the plant constant throughout its life.

(3) The working medium is at relatively high internal pressure. So, specific volume is less and the dimensions of the compressors and turbines can be reduced and the maximum unit capacity can be increased.

(4) A working medium such as helium, argon, hydrogen, neon may be used which has very good physical properties compared to those of air. For example thermal conductivity of hydrogen is about 6.8 times that of air and therefore requires smaller heat exchangers.

(5) The closed cycle can be operated with highest efficiencies in comparison to open cycle plants at an equal initial temperature of working fluid.

(6) The power output at constant speed can be varied by adding or substracting the working fluid and thus altering the weight of the charge. This gives improved part load efficiency as compared to open cycle gas turbine.

(7) High heat transfer can be possible.

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Disadvantages of closed cycle : **[May '12]**

(1) In closed cycle gas turbine considerable quantity of cooling water is required.
(2) Heat addition to the working medium takes place through heat exchanger, hence full heat of fuel fired is not utilized.
(3) Additional equipments such as externally fired heater, cooler, are required, the plant and its operations make additional complexity.

The closed cycle gas turbine would be preferable over open cycle one only when inferior type of fuel or solid fuel is to be used and cooling water at site is available.

9.8 Comparison between the open cycle and closed cycle gas turbines

Sr No	Particulars	Open cycle gas turbine	Closed cycle gas turbine
1.	Operation of cycle	The same working fluid is not recirculated again and again. The fresh charge is supplied to each cycle and after combustion and expansion it is discharged to the atm.	The working fluid is recirculated again and again.
2.	Working fluid	Air-fuel mixture	The working fluid air or any other suitable gas as Argon, Helium, Helium-CO_2 mixture. etc., can be used, which has more favourable properties than air.
3.	Manner of heat transfer	Fuel is directly mixed with air, direct heat supply and heat is generated in the combustion chamber itself.	Fuel is not mixed with working medium, and heat produced by fuel is transferred to working fluid through heat exchanger.
4.	Quality of heat input	It requires high grade heat energy	The heat can be supplied from any source like low grade fuel, waste heat, solar heat, nuclear heat etc.
5.	Type of fuel	It requires high quality oil or gaseous fuel because combustion is integral part of the system.	Any type of fuel can be used because heat is transfer externally
6.	Efficiency	Lower thermal efficiency for given temperature limits.	High thermal efficiency for given temperature limits.

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7.	Part load performance	Comparatively less thermal efficiency	Comparatively better thermal efficiency
8.	Compactness	Size of plant is large	Size of plant is less
9.	Control	Poor control on power production	Better control on power production
10.	Life	The combustion products directly passes through the turbine, and hence blades are subjected to higher thermal stresses and fouling. Therefore life of blade is shorter	Due to indirect heat transfer, combustion products do not come in contact of turbine blade, thus there is no blade fouling and longer blade life
11.	Complexity	It do not require heater and cooler and hence less complex	It requires heater and cooler and hence plant is more complex large
12.	Cost	Less	Large

9.9 Gas turbine fuel

The gas turbine engine can be operate with gases, liquid and solid fuels. However this statement is not true for all type of gas turbine engine. The selection of fuel for gas turbine is dependent on type of cycle, size and length of engine and application of gas turbine engine. The most important requirement of a gas turbine fuel are (i) composition should provide proper combustion and rapid burning, (ii) the products of combustion should contain a minimum of matter which might get deposited on the turbine blades or the heat exchanger, (iii) highest heating value per kg or m^3 volume specially for aircraft engines. The modern gas turbine are normally operated on the following fuels :

1. Gaseous fuels : Natural gas, Blast furnace gas, Producer gas, Sewage gas etc.
2. Liquid fuels : Kerosene, Gasoline, Residual oils etc.
3. Solid fuels : Coal

The solid fuel (coal) is normally used in closed cycle gas turbine. The main problem with using solid fuels in the case of open cycle plants is of reducing the level of flyash in the gases leaving the combustion chamber so that the blades of the turbines are not eroded. For this, it is necessary to ensure proper cleaning of the combustion gases from fine ash and dust. Before the gas enters the turbine, it is purified with help of the various fly ash collectors. The use of coal as a fuel of open cycle gas turbine is under the development. Coal is burnt in two methods. In the first method the coal is completely or partially gasified and fuel gas produced is supplied to gas turbine combustor. In other method, pressurized bubbling or circulating fluidized bed, where fuel gas, after it is adequately filtered, expands in the gas turbine. Coal is usually considered as a gas turbine fuel in combined cycle power plant.

Liquid oil line kerosene, gasoline etc. are easily used in a gas turbine combustor. Residual oils can be used as a fuel in gas turbine. If the viscosity of this oil is high, some heating arrangement required.

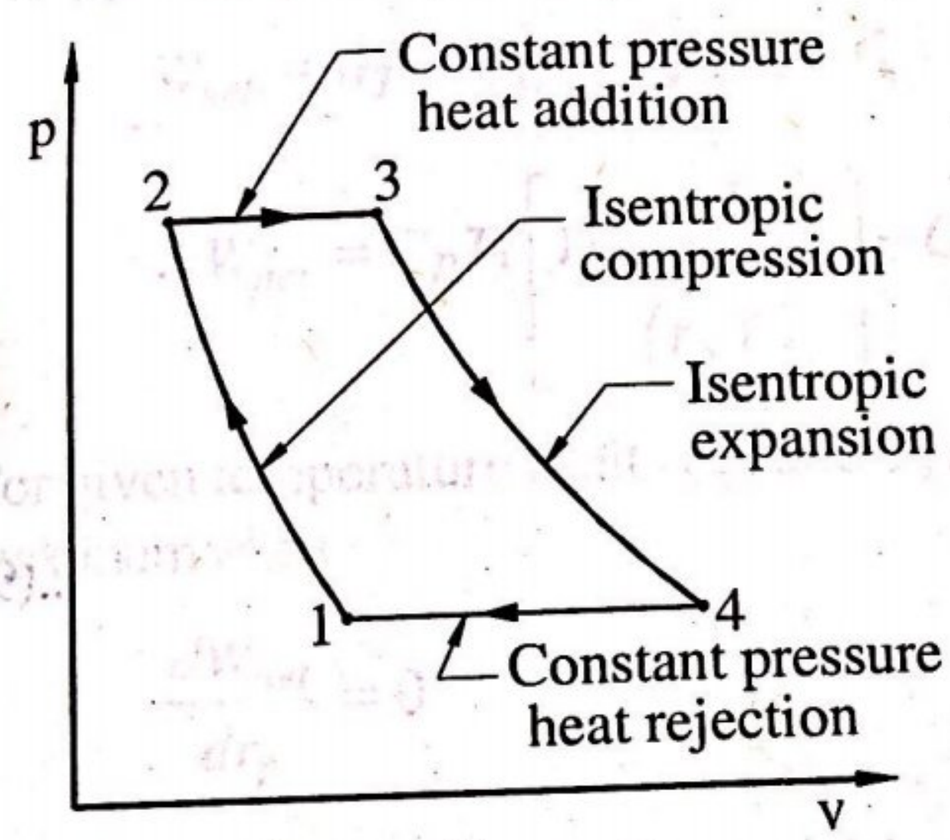
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9.10 Ideal Brayton cycle

[May '15]

The ideal cycle for gas turbine is Brayton or Jule cycle. The Brayton cycle has two constant pressure and two isentropic processes. Fig.9.4 shows p-V and T-S diagram of Brayton cycle. The air enters the compressor at point 1 and is compressed isentropically to point 2. The air then enters the combustion chamber where fuel is injected into it and heat is added to the air at constant pressure to raise its temperature from point 2 to point 3. The high pressure and temperature gases coming from combustion chamber enters gas turbine at point 3 and is expanded isentropically to point 4.



(a) p - v diagram *(b) T - s diagram*

Fig. 9.4 Brayton cycle

Process : 1-2 Isentropic compression
Process : 2-3 Constant pressure heat addition in combustion chamber
Process : 3-4 Isentropic expansion

Analysis of Brayton cycle :

[May '14]

The analysis is carried out with following assumptions :

(i) Compression and expansion are isentropic.
(ii) Pressure losses in the system are neglected.
(iii) Heat losses in the system are neglected.
(iv) Specific heat of working fluid is taken constant throughout the cycle.

Considering 1 kg of working fluid,
Work done by the compressor

$$W_C = C_P(T_2 - T_1)$$

Work produced by the turbine

$$W_T = C_P(T_3 - T_4)$$

Net work available for electric generation

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$$W_{net} = W_T - W_C$$
$$= C_P(T_3 - T_4) - C_P(T_2 - T_1)$$

Heat supplied in the combustion chamber

$$Q_S = C_P(T_3 - T_2)$$

Thermal efficiency of cycle

$$\eta_{th} = \frac{\text{Net work output}}{\text{Heat supplied}} = \frac{W_{net}}{Q_S} = \frac{C_P(T_3 - T_4) - C_P(T_2 - T_1)}{C_P(T_3 - T_2)}$$

$$= \frac{(T_3 - T_4) - (T_2 - T_1)}{(T_3 - T_2)}$$

$$= \frac{(T_3 - T_2) - (T_4 - T_1)}{(T_3 - T_2)}$$

$$\therefore \eta_{th} = 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)}$$

For process 1 – 2, $\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(r_p\right)^{\frac{\gamma-1}{\gamma}}$

$$\therefore T_2 = T_1\left(r_p\right)^{\frac{\gamma-1}{\gamma}}$$

where r_p = pressure ratio

For process 3 – 4, $\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(r_p\right)^{\frac{\gamma-1}{\gamma}}$

$$\therefore T_3 = T_4\left(r_p\right)^{\frac{\gamma-1}{\gamma}}$$

$$\therefore \eta_{th} = 1 - \frac{T_4 - T_1}{T_4\left(r_p\right)^{\frac{\gamma-1}{\gamma}} - T_1\left(r_p\right)^{\frac{\gamma-1}{\gamma}}}$$

$$= 1 - \frac{(T_4 - T_1)}{(\quad)^{\frac{\gamma-1}{\gamma}}}$$

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$$\therefore \eta_{th} = 1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}} \qquad \text{[May '14]} \qquad ...(9.2)$$

The above equation shows that the efficiency of simple gas turbine cycle is dependent only on the pressure ratio. As the pressure ratio increases the efficiency of the turbine increases. The efficiency is independent on the turbine inlet temperature.

Condition for maximum work :

The net work produced by gas turbine plant is given by

$$W_{net} = W_T - W_C = C_P(T_3 - T_4) - C_P(T_2 - T_1)$$

$$\therefore W_{net} = C_P T_3\left[1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}}\right] - C_P T_1\left[(r_p)^{\frac{\gamma-1}{\gamma}} - 1\right] \qquad ...(9.3)$$

For given temperature limits (T_1 and T_3 are fixed). The work produced by plant becomes maximum when

$$\frac{dW_{net}}{dr_p} = 0$$

$$\therefore \frac{d}{dr_p}\left[C_P T_3 - C_P T_3 (r_P)^{\frac{\gamma-1}{\gamma}} - C_P T_1 (r_P)^{\frac{\gamma-1}{\gamma}} + C_P T_1\right] = 0$$

$$\therefore 0 - C_P T_3\left[-\left(\frac{\gamma-1}{\gamma}\right)\right](r_P)^{-\left(\frac{\gamma-1}{\gamma}\right)-1} - C_P T_1 \frac{\gamma-1}{\gamma}(r_P)^{\frac{\gamma-1}{\gamma}-1} + 0 = 0$$

$$\therefore T_3 (r_P)^{\frac{-\gamma+1-\gamma}{\gamma}} - T_1 (r_P)^{\frac{\gamma-1-\gamma}{\gamma}} = 0$$

$$\therefore T_3 (r_P)^{\frac{-2\gamma+1}{\gamma}} - T_1 (r_P)^{\frac{1}{\gamma}} = 0$$

$$\therefore \frac{T_3}{T_1} = \frac{(r_P)^{\frac{1}{\gamma}}}{(r_P)^{\frac{-2\gamma+1}{\gamma}}}$$

$$\therefore \frac{T_3}{T_1} = (r_P)^{\frac{2(\gamma-1)}{\gamma}}$$

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$$\therefore r_P = \left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}} \quad \ldots(9.4)$$

The above equation shows that the pressure ratio for maximum work is a function of the limiting temperature ratio.

Maximum work :

The condition of maximum work $r_P = \left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}}$ putting into equation (9.3) we get

$$(W_{net})_{max} = C_P T_3 \left[1 - \frac{1}{\left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)} \times \frac{\gamma-1}{\gamma}}}\right] - C_P T_1 \left[\left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)} \times \frac{\gamma-1}{\gamma}} - 1\right]$$

$$= C_P T_3 \left[1 - \frac{1}{\left(\frac{T_3}{T_1}\right)^{\frac{1}{2}}}\right] - C_P T_1 \left[\left(\frac{T_3}{T_1}\right)^{\frac{1}{2}} - 1\right]$$

$$= C_P T_3 \left[1 - \frac{\sqrt{T_1}}{\sqrt{T_3}}\right] - C_P T_1 \left[\frac{\sqrt{T_3}}{\sqrt{T_1}} - 1\right]$$

$$= C_P T_3 \left[\frac{\sqrt{T_3} - \sqrt{T_1}}{\sqrt{T_3}}\right] - C_P T_1 \left[\frac{\sqrt{T_3} - \sqrt{T_1}}{\sqrt{T_1}}\right]$$

$$= C_P \sqrt{T_3}\left[\sqrt{T_3} - \sqrt{T_1}\right] - C_P \sqrt{T_1}\left[\sqrt{T_3} - \sqrt{T_1}\right]$$

$$= C_P \left[T_3 - \sqrt{T_1 T_3}\right] - C_P \left[\sqrt{T_1 T_3} - T_1\right]$$

$$\therefore (W_{net})_{max} = C_P \left[T_1 + T_3 - 2\sqrt{T_1 T_3}\right] \quad \ldots(9.5)$$

The above equation shows that the maximum work is dependent on the limiting mperature range (T_1 and T_3).

Now, considering the masses of air and fuel passing through the open cycle gas turbine.

Let, m_a = mass of air supplied to combustion chamber, kg/sec.

C_{pg} = specific heat of gases

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C_{pa} = specific heat of air

$$\therefore W_T = (m_a + m_f)C_{pg}(T_3 - T_4)$$

$$W_C = m_a C_{pa}(T_2 - T_1)$$

$$W_{net} = (m_a + m_f)C_{pg}(T_3 - T_4) - m_a C_{pa}(T_2 - T_1)$$

Heat supplied to the combustion chamber

$$Q_s = m_f \times CV \times \eta_{comb}$$

where CV = Calorific Value of fuel

η_{comb} = combustion efficiency

Neglecting heat losses in the combustion chamber i.e. $\eta_{comb} = 100\%$

$$\therefore Q_s = m_f \cdot C.V.$$

$$\therefore \eta_{th} = \frac{W_{net}}{Q_s} = \frac{(m_a + m_f)C_{pg}(T_3 - T_4) - m_a C_{pa}(T_2 - T_1)}{m_f \times CV} \quad ...(9.6)$$

9.11 Work ratio and air rate

Work ratio : *It is the ratio of net work produced by plant to the turbine work.* The work ratio represents the amount of net work generated by the gas turbine plant per unit turbine work output.

$$\text{Work ratio} = \frac{W_{net}}{W_T} = \frac{W_T - W_C}{W_T} = 1 - \frac{W_C}{W_T}$$

$$= 1 - \frac{T_2 - T_1}{T_3 - T_4} = 1 - \frac{T_1 (r_p)^{\frac{\gamma-1}{\gamma}} - T_1}{T_3 - \frac{T_3}{(r_p)^{\frac{\gamma-1}{\gamma}}}}$$

$$WR = 1 - \frac{T_1}{T_3}(r_p)^{\frac{\gamma-1}{\gamma}} \quad ...(9.7)$$

The work ratio increases with increase in turbine inlet temperature, decrease in compressor inlet temperature and decrease in pressure ratio of cycle. The compressor inlet temperature is always atmospheric temperature particularly in open cycle and turbine inlet temperature is limited by metallurgical considerations. The highest temperature ever used for gas turbine power plants is about 1000 K.

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Air rate : The size of gas turbine plant is dependent upon the rate of air flow in relation to net output work. Compact gas turbine plant has a lower value of air rate. Air rate is expressed as kg of air per kWhr.

Air rate is defined as :

$$AR = \frac{\text{kg of air taken/hr}}{\text{power output in kW}} = \frac{m_a \times 3600}{m_a \times net\ work\ in\ kJ/kg} = \frac{3600}{W_T - W_C}\ kg/kW\ hr \quad ...(9.8)$$

where, m_a = mass flow rate of air, kg/s

Problem 9.1 *In a constant pressure open cycle gas turbine air enters the compressor at 1 bar and 18°C where it is compressed to a pressure ratio of 6. The gases enters the gas turbine at 730°C and expands to original pressure. Calculate the work ratio and the thermal efficiency when gas turbine plant operates on a Brayton cycle.*

Assume, $\gamma = 1.4, C_P = 1.0\ kJ/kg\ K$ *for air and* $\gamma = 1.3, C_P = 1.1\ kJ/kg\ K$ *for gases. Neglect the mass of fuel.*

Solution: Given data : Refer Fig 9.5 ideal Brayton cycle on T-S diagram,

$$p_1 = 1\ bar,\ \frac{p_2}{p_1} = 6,\ T_1 = 18°C = 291K,\ T_3 = 730°C = 1003K,\ WR = ?,\ \eta_{th} = ?$$

For isentropic process 1-2

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \quad \therefore T_2 = 291 \times (6)^{\frac{1.4-1}{1.4}} \quad \therefore T_2 = 485.53K$$

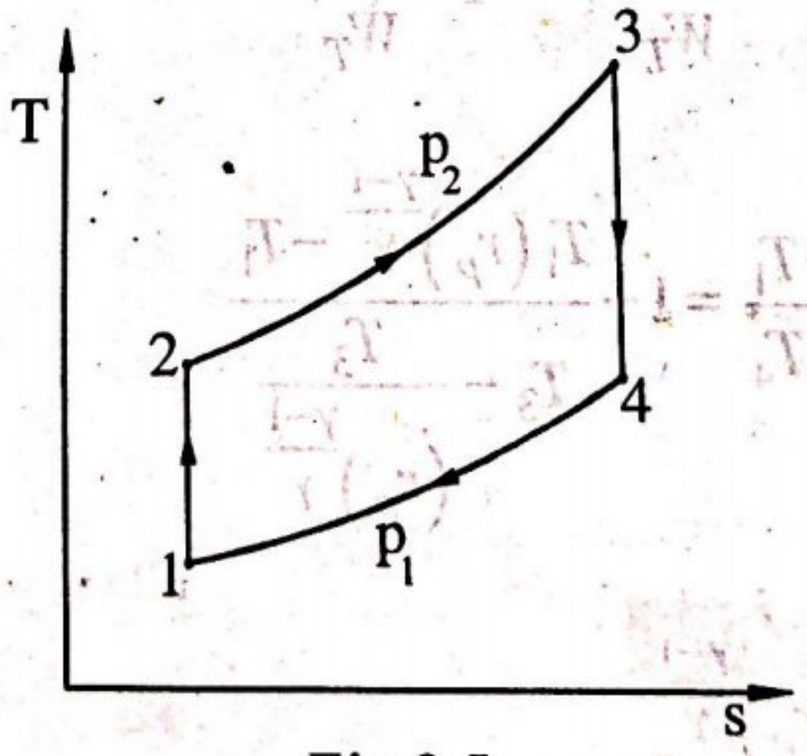


Fig.9.5

For isentropic process 3-4

$$\frac{T_3}{T_4} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \quad \therefore T_4 = 1003\left(\frac{1}{6}\right)^{\frac{1.3-1}{1.3}} = 663.32K$$

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Turbine work output

$$W_T = C_{pg}(T_3 - T_4)\,KJ/kg\ of\ air$$
$$= 1.1\times(1003 - 663.32) = 373.67 kJ/kg$$

Compressor work input

$$W_C = C_{pa}(T_2 - T_1)\,KJ/kg\ of\ air$$
$$= 1\times(485.53 - 291) = 194.53 kJ/kg$$

Net work output

$$W_{net} = W_T - W_C = 373.67 - 194.53 = 179.11 kJ/kg$$

Work ratio $WR = \dfrac{W_{net}}{W_T} = \dfrac{179.11}{373.67} = 0.479$

Heat supplied $Q_S = C_{pg}(T_3 - T_2) = 1.1\times(1003 - 485.53) = 569.217 kJ/kg$

Thermal efficiency $\eta_{th} = \dfrac{W_{net}}{Q_S} = \dfrac{179.11}{569.217} = 0.3146g = \mathbf{31.46\%}$ ***Ans***

Problem 9.1a *A gas turbine installations works on Brayton cycle between the temperture limits of 35°C and 715°C. For the maximum work to be developed, calculate temperature at the end of compression, pressure ratio and thermal efficiency.* ***[May '14]***

Solution: Given data :

$T_1 = 35°C = 308\,K,\ T_3 = 715°C = 988K, T_2 = ?, r_p = ?, \eta_{th} = ?$

For maximum work, the presure ratio is given by

$$r_p = \left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}} = \left(\frac{988}{308}\right)^{\frac{1.4}{2(1.4-1)}} = \mathbf{7.688}$$ ***Ans***

For isentropic compression process 1–2,

$$\frac{T_2}{T_1} = (r_p)^{\frac{\gamma-1}{\gamma}} \quad \therefore T_2 = 308(7.688)^{\frac{1.4-1}{1.4}} = \mathbf{551.63\,K}$$ ***Ans***

Thermal efficiency

$$\eta_{th} = 1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}} = 1 - \frac{1}{(7.688)^{\frac{1.4-1}{1.4}}} = \mathbf{44.16\%}$$ ***Ans***

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9.12 Actual Brayton cycle

The actual gas turbine (Brayton) cycle differs from an ideal Brayton cycle in the following manner.

(1) The working medium does not follow the ideal gas low.

(2) Due to frictional losses in the compressor and turbine, the compression and expansion processes are not frictionless which makes the processes irreversible. In ideal case, the compression and expansion processes is considered isentropic (compressor and turbine efficiencies are 100% each), but actual compressor and turbine have efficiency less than 100%.

(3) In actual cycle, there are pressure losses in flow through ducting connecting the various components particularly in the combustion chamber and heat exchanger.

(4) The specific heats of working fluid will vary throughout the cycle due to temperature change and mixing with combustion gases. The specific heat increases with an increases in temperature. This causes lower temperature rise for a given heat input.

(5) The fluid velocities are high in turbo-machinery, hence the change in kinetic energy between inlet and outlet of each component cannot necessarily be ignored.

(6) Actual work output is less than the ideal work output by an amount equal to friction, windage and transmission losses which cannot be ignored.

(7) The mass of the gas flowing through the turbine is (1 + F/A) times the mass of air flowing through the compressor. (F/A = Fuel air ratio).

(8) In order to avoid back flow at the exhaust of gas turbine, the actual gas pressure at outlet of the turbine is greater than the atmospheric pressure. Therefore, there is pressure loss in exhaust hood of turbine.

The above factors, which tend to reduce the efficiency and power output of the actual cycle can be taken into consideration in the calculation of its performance by assigning efficiency values of each one of them. The deviation from ideal compression is taken into account by efficiency of compressor η_c and deviation from ideal expansion is taken into account by efficiency of turbine η_t. The effect of deviation of working medium from ideal gas law can be considered by taking into account the effect of change in ratio of specific heats from the γ to polytropic index n. The change in kinetic energy between inlet and outlet of each component can be account by considering stagnation values of properties instead of static values. Normally, for the sake of simplicity of the problem, the pressure drop in combustion chamber is ignored.

Fig. 9.6 shows an actual Brayton cycle on T-s diagram

Process 1-2 : Ideal (isentropic) compression

1-2′ : Actual compression

Process 3-4 : Ideal (isentropic) expansion

3-4′ : Actual expansion

Pressure loss in the combustion chamber = $p_2 - p_3$

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Pressure loss in the exhaust hood of turbine = $p_4 - p_1$

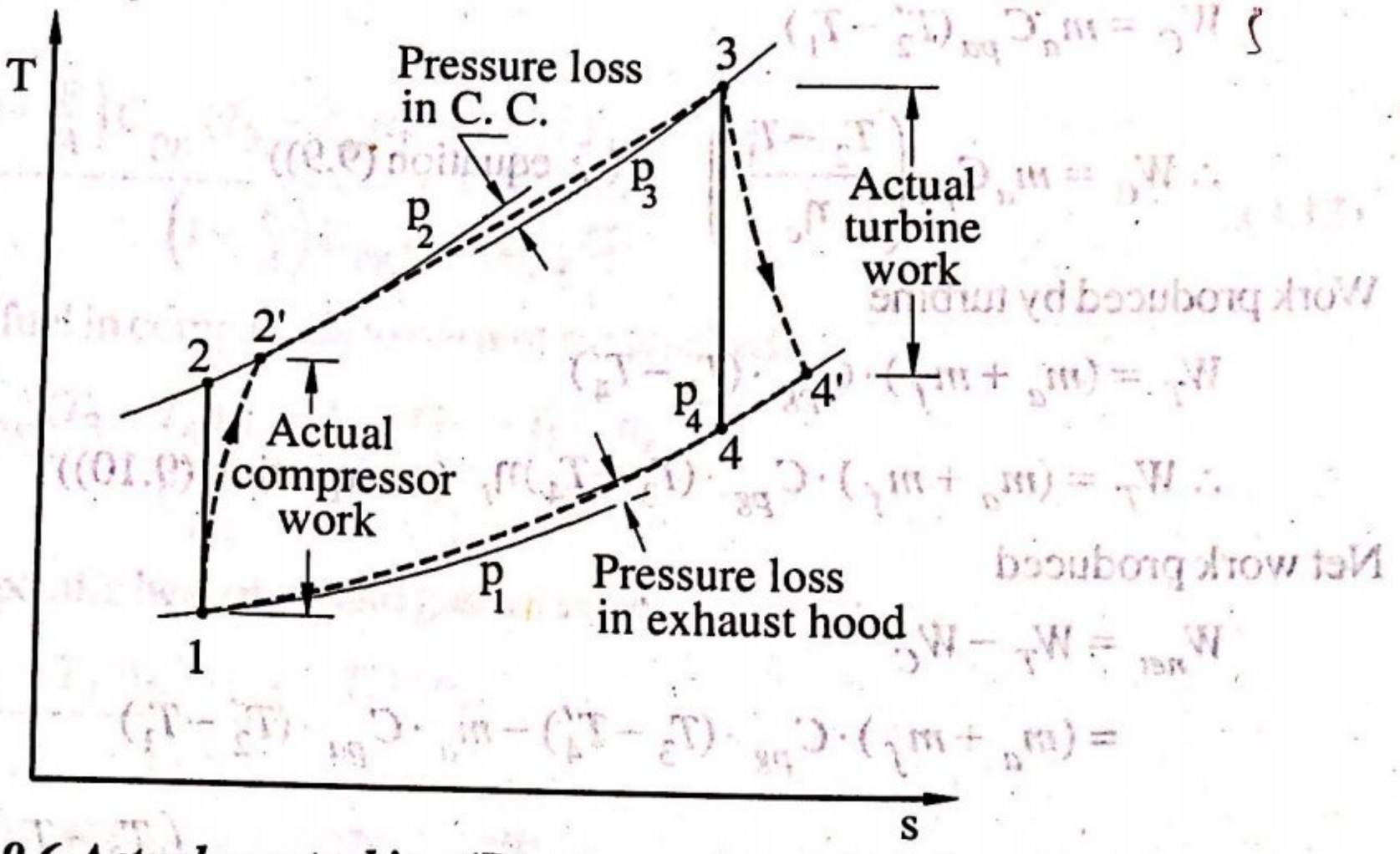


Fig. 9.6 Actual gas turbine (Brayton cycle) on T-s diagram

Compressor and turbine efficiency : During the compression process part of energy is wasted due to friction in irreversible process. The result is that the temperature of working fluid is higher at the end of compression compared to that of ideal process. Hence, some extra work is required to the compressor because of its inefficiency. Similarly, during the expansion process, the exhaust gas temperature is higher compared to that of ideal expansion. Hence, turbine produces less work compared to that of ideal cycle.

The compressor efficiency can be defined as ratio of isentropic compression work to actual compression work.

$$\eta_c = \frac{\text{Isentropic compression work}}{\text{Actual compression work}} = \frac{C_{pa}(T_2 - T_1)}{C_{pa}(T_2' - T_1)}$$

$$\therefore \boxed{\eta_c = \frac{T_2 - T_1}{T_2' - T_1}} \quad \ldots(9.9)$$

The turbine efficiency can be defined as the ratio of actual turbine work to isentropic turbine work

$$\eta_t = \frac{\text{Actual turbine work}}{\text{Isentropic turbine work}} = \frac{C_{pg}(T_3 - T_4')}{C_{pg}(T_3 - T_4)}$$

$$\therefore \boxed{\eta_t = \frac{T_3 - T_4'}{T_3 - T_4}} \quad \ldots(9.10)$$

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Thermal efficiency of cycle :

Work required for compressor

$$W_C = m_a C_{pa}(T_2' - T_1)$$

$$\therefore W_C = m_a C_{pa}\left(\frac{T_2 - T_1}{\eta_c}\right) \quad (\because \text{ equation (9.9)})$$

Work produced by turbine

$$W_T = (m_a + m_f)\cdot C_{pg}\cdot(T_3 - T_4')$$

$$\therefore W_T = (m_a + m_f)\cdot C_{pg}\cdot(T_3 - T_4)\eta_t \quad (\because \text{ equation (9.10)})$$

Net work produced

$$W_{net} = W_T - W_C$$

$$= (m_a + m_f)\cdot C_{pg}\cdot(T_3 - T_4') - m_a\cdot C_{pa}\cdot(T_2' - T_1)$$

$$\therefore W_{net} = (m_a + m_f)\cdot C_{pg}\cdot(T_3 - T_4)\eta_t - m_a\cdot C_{pa}\cdot\left(\frac{T_2 - T_1}{\eta_c}\right)$$

$$= m_a\left(1 + \frac{m_f}{m_a}\right)C_{pg}(T_3 - T_4)\cdot\eta_t - m_a C_{pa}(T_2 - T_1)\times\frac{1}{\eta_c}, \text{ kJ/s}$$

$$= \left(1 + \frac{m_f}{m_a}\right)C_{pg}(T_3 - T_4)\cdot\eta_t - C_{pa}(T_2 - T_1)\times\frac{1}{\eta_c}, \text{ kJ/kg of air}$$

$$\therefore W_{net} = \left(1 + \frac{F}{A}\right)\cdot C_{pg}(T_3 - T_4)\cdot\eta_t - C_{pa}(T_2 - T_1)\times\frac{1}{\eta_c} \quad ...(9.11)$$

$$\text{where } \frac{F}{A} = \frac{m_f}{m_a} = \text{fuel air ratio}$$

Heat supplied in combustion chamber

$$Q_s = (m_a + m_f)C_{pg}T_3 - m_a C_{pa}T_2'$$

$$= m_a\left[\left(1 + \frac{m_f}{m_a}\right)\cdot C_{pg}T_3 - C_{pa}T_2'\right], \text{ kJ/s}$$

$$\therefore Q_s = \left(1 + \frac{F}{A}\right)C_{pg}T_3 - C_{pa}T_2', \text{ kJ/kg of air} \quad ...(9.12)$$

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Thermal efficiency of cycle $(\eta_{th}) = \frac{W_{net}}{Q_s}$

$$\therefore \eta_{th} = \frac{\left(1+\frac{F}{A}\right)C_{pg}(T_3 - T_4)\eta_t - C_{pa}(T_2 - T_1)\times\frac{1}{\eta_c}}{\left(1+\frac{F}{A}\right)C_{pg}T_3 - C_{pa}T_2'} \quad ...(9.13)$$

Neglecting mass of fuel in comparison to mass of air supplied

$$\therefore \eta_{th} = \frac{C_{pg}(T_3 - T_4)\eta_t - C_{pa}(T_2 - T_1)/\eta_c}{C_{pg}T_3 - C_{pa}T_2'}$$

Assuming that the specific heat of air and gas are same

$$\therefore \eta_{th} = \frac{(T_3 - T_4)\eta_t - (T_2 - T_1)/\eta_c}{T_3 - T_2'}$$

$$\therefore \eta_{th} = \frac{(T_3 - T_4)\cdot\eta_t\cdot\eta_c - (T_2 - T_1)}{(T_3 - T_2')\eta_c} \quad ...(9.14)$$

From above equation it is clear that the thermal efficiency of gas turbine plant depends on turbine inlet temperature as well as compressor and turbine efficiency. The thermal efficiency is increased by improving either compressor or turbine efficiency or both. However, it should be remember that the additional work-supplied to the compressor because of its inefficiency is not entirely wasteful because as compared to isentropic compression, the heat supplied in the combustion chamber is reduced.

9.13 Optimum pressure ratio for maximum work output in actual cycle

[May '12]

The net work produced by gas turbine is given by

$$W_{net} = \left(1+\frac{F}{A}\right)\cdot C_{pg}(T_3 - T_4)\cdot\eta_t - C_{pa}(T_2 - T_1)\times\frac{1}{\eta_c}$$

$$\therefore W_{net} = \left(1+\frac{F}{A}\right)\cdot C_{pg}T_3\left(1-\frac{T_4}{T_3}\right)\cdot\eta_t - C_{pa}T_1\left(\frac{T_2}{T_1}-1\right)\cdot\frac{1}{\eta_c}$$

$$= \left(1+\frac{F}{A}\right)\cdot C_{pg}\cdot T_3\left[1-\frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}}\right]\cdot\eta_t - C_{pa}T_1\left[(r_p)^{\frac{\gamma-1}{\gamma}}-1\right]\times\frac{1}{\eta_c}$$

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$$\left(\because \frac{T_3}{T_4} = \frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}}\right)$$

Neglecting mass of fuel and assuming $C_{pa} = C_{pg} = C_p$ we get

$$W_{net} = C_p T_3 \left[1 - (r_p)^{-\left(\frac{\gamma-1}{\gamma}\right)}\right] \cdot \eta_t - \frac{C_p T_1}{\eta_c}\left[(r_p)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

For given value of turbine, compressor efficiencies (η_t, η_c) and turbine inlet temperature (T_3), the optimum pressure ratio for maximum work output can be obtained by

$$\frac{dW_{net}}{dr_p} = 0$$

$$\therefore \frac{d}{dr_p}\left\{C_p T_3 \left[1 - (r_p)^{-\left(\frac{\gamma-1}{\gamma}\right)}\right] \times \eta_t - \frac{C_p T_1}{\eta_c}\left[(r_p)^{\frac{\gamma-1}{\gamma}} - 1\right]\right\} = 0$$

$$\therefore C_p T_3 \eta_t \left[0 - \left(-\frac{\gamma-1}{\gamma}\right)(r_p)^{-\left(\frac{\gamma-1}{\gamma}\right)-1}\right] - \frac{C_p T_1}{\eta_c}\left[\left(\frac{\gamma-1}{\gamma}\right)(r_p)^{-\left(\frac{\gamma-1}{\gamma}\right)-1} - 0\right] = 0$$

$$\therefore T_3 \eta_t \left[\left(\frac{\gamma-1}{\gamma}\right)(r_p)^{\frac{-\gamma+1-\gamma}{\gamma}}\right] - \frac{T_1}{\eta_c}\left[\left(\frac{\gamma-1}{\gamma}\right)(r_p)^{\frac{\gamma-1-\gamma}{\gamma}}\right] = 0$$

$$\therefore T_3 \eta_t \eta_c (r_p)^{\frac{-2\gamma+1}{\gamma}} - T_1 (r_p)^{-\frac{1}{\gamma}} = 0$$

$$\therefore \frac{(r_p)^{\frac{-2\gamma+1}{\gamma}}}{(r_p)^{-\frac{1}{\gamma}}} = \frac{T_1}{T_3} \times \frac{1}{\eta_c \eta_t}$$

$$\therefore (r_p)^{-\frac{2(\gamma-1)}{\gamma}} = \frac{T_1}{T_3} \times \frac{1}{\eta_c \eta_t}$$

$$\therefore r_p = \left[\frac{T_3}{T_1} \cdot \eta_c \cdot \eta_t\right]^{\frac{\gamma}{2(\gamma-1)}} \quad \ldots(9.15)$$

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For ideal cycle $\eta_c = \eta_t = 1$

$$\therefore r_p = \left[\frac{T_3}{T_1}\right]^{\frac{\gamma}{2(\gamma-1)}}$$

9.14 Optimum pressure ratio for maximum thermal efficiency in actual cycle

The thermal efficiency of actual gas turbine cycle is given by

$$\eta_{th} = \frac{(T_3 - T_4)\cdot\eta_t\cdot\eta_c - (T_2 - T_1)}{(T_3 - T_2')\eta_c}$$

For simplification of problem, taking $(T_3 - T_2') \cong (T_3 - T_2)$

$$\therefore \eta_{th} = \frac{(T_3 - T_4)\cdot\eta_t\cdot\eta_c - (T_2 - T_1)}{(T_3 - T_2)\eta_c}$$

$$= \frac{T_3\left(1 - \frac{T_4}{T_3}\right)\eta_t\eta_c - T_1\left(\frac{T_2}{T_1} - 1\right)}{T_1\left(\frac{T_3}{T_1} - \frac{T_2}{T_1}\right)}$$

But $\frac{T_2}{T_1} = \frac{T_3}{T_4} = (r_p)^{\frac{\gamma-1}{\gamma}}$

$$\therefore \eta_{th} = \frac{T_3\left[1 - (r_p)^{-\frac{\gamma-1}{\gamma}}\right]\eta_t\cdot\eta_c - T_1\left[(r_p)^{\frac{\gamma-1}{\gamma}} - 1\right]}{T_1\left[\frac{T_3}{T_1} - (r_p)^{\frac{\gamma-1}{\gamma}}\right]}$$

For simplicity, $(r_p)^{\frac{\gamma-1}{\gamma}} = x$ (say) and $\frac{T_3}{T_1} = y$ (say)

$$\therefore \eta_{th} = \frac{T_3\left[1 - \frac{1}{x}\right]\eta_t\cdot\eta_c - T_1[x-1]}{T_1[y-x]}$$

For given value of T_3, T_1, η_c and η_t, the optimum pressure ratio for maximum efficiency can be obtained by

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$$\frac{d\eta_{th}}{dr_p} = 0$$

$$\therefore \frac{d\eta_{th}}{dx} \cdot \frac{dx}{dr_p} = 0$$

But $x = \left(r_p\right)^{\frac{\gamma-1}{\gamma}} \quad \therefore dx = \frac{\gamma-1}{\gamma}(r_p)^{\frac{\gamma-1}{\gamma}-1} dr_p$

$$\therefore dx = \frac{\gamma-1}{\gamma}(r_p)^{\frac{-1}{\gamma}} dr_p \qquad \therefore \frac{dx}{dr_p} = \frac{\gamma-1}{\gamma}(r_p)^{\frac{-1}{\gamma}}$$

$$\therefore \frac{d\eta_{th}}{dx} \cdot \frac{\gamma-1}{\gamma}(r_p)^{\frac{-1}{\gamma}} = 0 \qquad \therefore \frac{d\eta_{th}}{dx} = 0$$

$$\therefore \frac{d}{dx}\left\{\frac{T_3\left[1-\frac{1}{x}\right]\eta_t \cdot \eta_c - T_1[x-1]}{T_1[y-x]}\right\} = 0$$

$$\therefore \frac{d}{dx}\left\{\frac{T_3}{T_1}\frac{\left(1-\frac{1}{x}\right)\eta_t \cdot \eta_c}{(y-x)}\right\} - \frac{d}{dx}\left\{\frac{x-1}{y-x}\right\} = 0$$

$$\therefore y\eta_t \cdot \eta_c \frac{d}{dx}\left\{\frac{1-\frac{1}{x}}{y-x}\right\} - \frac{d}{dx}\left\{\frac{x-1}{y-x}\right\} = 0 \quad \left(\because \frac{T_3}{T_1} \equiv y\right)$$

$$\therefore y\eta_t \cdot \eta_c \left\{\frac{\left(0+\frac{1}{x^2}\right)(y-x)-\left(1-\frac{1}{x}\right)(0-1)}{(y-x)^2}\right\} - \left\{\frac{(1-0)(y-x)-(x-1)(0-1)}{(y-x)^2}\right\} = 0$$

$$\therefore y\eta_c\eta_t \left\{\frac{y-x}{x^2}+1-\frac{1}{x}\right\} - \{y-x+x-1\} = 0$$

$$\therefore \frac{\eta_c\eta_t y^2}{x^2} - \frac{\eta_c\eta_t y}{x} + \eta_c\eta_t y - \frac{\eta_c\eta_t y}{x} - y + 1 = 0$$

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$$\therefore \frac{1}{x^2}(\eta_c \eta_t y^2) - \frac{1}{x}(2\eta_c \eta_t y) + [y(\eta_c \eta_t - 1) + 1] = 0$$

Assuming $\frac{1}{x} = z$

$$\therefore z^2(\eta_c \eta_t y^2) - z(2\eta_c \eta_t y) + [y(2\eta_c \eta_t - 1) + 1] = 0$$

Above equation is quadratic equation in z, as

$$az^2 + bz + c = 0$$

$$\therefore a = \eta_c \eta_t y^2, b = -2\eta_c \eta_t y, c = y(\eta_c \eta_t - 1) + 1$$

Solution of this equation is given by

$$z = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}$$

Taking positive roots we get

$$z = \frac{1}{x} = \frac{+2\eta_c \eta_t y + \sqrt{4(\eta_c \eta_t y)^2 - 4\eta_c \eta_t y^2 [y(\eta_c \eta_t - 1) + 1]}}{2\eta_c \eta_t y^2}$$

$$= \frac{1 + \sqrt{1 - \left[y\left(1 - \frac{1}{\eta_c \eta_t}\right) + \frac{1}{\eta_c \eta_t} \right]}}{y}$$

$$= \frac{1 + \sqrt{1 - y + \frac{y}{\eta_c \eta_t} - \frac{1}{\eta_c \eta_t}}}{y}$$

$$= \frac{1 + \sqrt{-(y-1) + \frac{1}{\eta_c \eta_t}(y-1)}}{y}$$

$$= \frac{1 + \sqrt{(y-1)\left(\frac{1}{\eta_c \eta_t}\right) - (y-1)}}{y}$$

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$$\therefore \frac{1}{x} = \frac{1+\sqrt{(y-1)\left(\frac{1}{\eta_c \eta_t}-1\right)}}{y}$$

$$\therefore x = \frac{y}{1+\sqrt{(y-1)\left(\frac{1}{\eta_c \eta_t}-1\right)}}$$

$$\therefore (r_p)^{\frac{\gamma-1}{\gamma}} = \frac{\frac{T_3}{T_1}}{1+\sqrt{\left(\frac{T_3}{T_1}-1\right)\left(\frac{1}{\eta_c \eta_t}-1\right)}} \qquad \left(\because x = (r_p)^{\frac{\gamma-1}{\gamma}},\ and\ y = \frac{T_3}{T_1}\right)$$

$$\therefore (r_p)_{opt} = \left[\frac{\frac{T_3}{T_1}}{1+\sqrt{\left[\left(\frac{T_3}{T_1}-1\right)\left(\frac{1}{\eta_c \eta_t}-1\right)\right]}}\right]^{\frac{\gamma}{\gamma-1}} \qquad ...(9.16)$$

Minimum temperature ratio in the gas turbine :

The work supplied to compressor and work produced by turbine is given by

$$W_c = m_a \cdot C_{pa}(T_2 - T_1)/\eta_c$$

$$W_T = (m_a + m_f) \cdot C_{pg}(T_3 - T_4)\eta_t$$

Neglecting mass flow of fuel, and assuming $C_{pa} = C_{pg}$, we get work per kg of air supplied

$$W_c = C_p(T_2 - T_1) \times \frac{1}{\eta_c} = C_p T_1\left(\frac{T_2}{T_1}-1\right) \times \frac{1}{\eta_c} = \frac{C_p T_1}{\eta_c}\left[(r_p)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

$$W_T = C_p(T_3 - T_4)\eta_t = C_p T_3\left(1-\frac{T_4}{T_3}\right)\eta_t = C_p T_3\left[1-\frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}}\right] \cdot \eta_t$$

If $W_C = W_T$, the minimum temperature ratio $\left(\frac{T_3}{T_1}\right)$ can be obtained for which no net work is possible.

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$$\therefore W_C = W_T$$

$$\therefore \frac{C_p T_1}{\eta_c}\left[(r_p)^{\frac{\gamma-1}{\gamma}} - 1\right] = C_p T_3 \left[1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}}\right] \cdot \eta_t$$

$$\therefore \left(\frac{T_3}{T_1}\right)_{\min} = \frac{1}{\eta_c \eta_t} \frac{\left[(r_p)^{\frac{\gamma-1}{\gamma}} - 1\right]}{\left[1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}}\right]} = \frac{1}{\eta_c \eta_t} \frac{\left[(r_p)^{\frac{\gamma-1}{\gamma}} - 1\right](r_p)^{\frac{\gamma-1}{\gamma}}}{\left[(r_p)^{\frac{\gamma-1}{\gamma}} - 1\right]}$$

$$\therefore \left(\frac{T_3}{T_1}\right)_{\min} = \frac{1}{\eta_c \eta_t}(r_p)^{\frac{\gamma-1}{\gamma}} \quad \ldots(9.17)$$

Problem 9.2 *The air enters the compressor of an open cycle constant pressure gas turbine at a pressure of 1 bar and temperature 20°C. The pressure of the air after compression is 4 bar. The isentropic efficiency of compressor and turbine are 80% and 85% respectively. The air fuel ratio used as 90:1, of the flow rate of air is 3kg per second. Find*

(i) Power developed

(ii) Thermal efficiency of the cycle

Assume, $C_P = 1\ kJ/kgK$ *and* $\gamma = 1.4$ *for air & gas and C.V of fuel = 41,800 kJ/kg.*

Solution: Given data: Refer Fig. 9.7

$p_1 = 1\ bar,\ T_1 = 20°C = 293K,\ p_2 = 4\ bar,\ \eta_c = 0.8,\ \eta_t = 0.85,\ \frac{m_a}{m_f} = 90:1,$

$m_a = 3\ kg/s,\ P = ?,\ \eta_{th} = ?$

For compression process 1-2

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \quad \therefore T_2 = 293 \times \left(\frac{4}{1}\right)^{\frac{1.4-1}{1.4}} = 435.39K$$

$$\text{Also, } \eta_c = 0.8 = \frac{T_2 - T_1}{T_2' - T_1}$$

$$\therefore T_2' = T_1 + \frac{T_2 - T_1}{0.8} \quad \therefore T_2' = 293 + \frac{435.39 - 293}{0.8} = 470.99K$$

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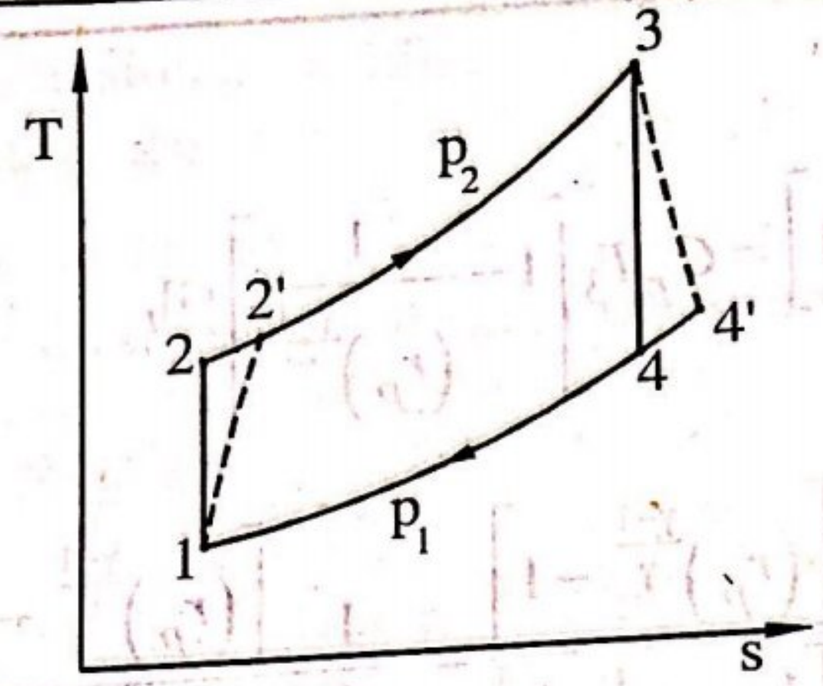


Fig. 9.7

Heat balance for combustion process $2'-3$:

Heat supplied to the combustion chamber = Heat gain by working fluid

$$\therefore m_f .CV \times \eta_{comb} = (m_a + m_f)C_P T_3 - m_a C_{p_2} T_2'$$

Assuming combustion efficiency $\eta_{comb} = 100\%$

$$\therefore m_f \times CV = (m_a + m_f)C_P T_3 - m_a C_p T_2'$$

$$\therefore CV = \left(\frac{m_a}{m_f} + 1\right) C_P T_3 - \frac{m_a}{m_f} C_p T_2'$$

$$\therefore 41800 = \left(\frac{90}{1} + 1\right) \times 1 \times T_3 - \frac{90}{1} \times 1 \times 470.99$$

$$\therefore T_3 = 925.15K$$

For expansion process 3-4

$$\therefore \frac{T_3}{T_4} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \quad \therefore T_4 = T_3 \times \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = 925.15 \times \left(\frac{1}{4}\right)^{\frac{1.4-1}{1.4}} = 622.57K$$

Also, $\eta_t = 0.85 = \dfrac{T_3 - T_4'}{T_3 - T_4}$

$$\therefore T_4' = T_3 - 0.85(T_3 - T_4) = 925.15 - 0.85(925.15 - 622.57) = 667.95K$$

Net work produced $W_{net} = W_T - W_C$

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$$\therefore W_{net} = (m_a + m_f) C_p (T_3 - T_4') - m_a C_p (T_2' - T_1)$$

$$= m_a \left(1 + \frac{m_f}{m_a}\right) C_p (T_3 - T_4') - m_a C_p (T_2' - T_1)$$

$$= 3 \times \left(1 + \frac{1}{90}\right) \times 1 \times (925.15 - 667.95) - 3 \times 1 \times (470.99 - 293)$$

$$= 780.15 - 533.97 = 246.18 \ kJ/\text{sec}$$

$$= \mathbf{246.18\ kW}$$ **Ans**

Thermal efficiency of cycle

$$\eta_{th} = \frac{W_{net}}{Q_s} = \frac{W_{net}}{m_f \times CV} = \frac{246.18}{\frac{3}{90} \times 41,800} \times 100 = \mathbf{17.66\%}$$ **Ans**

Problem 9.3 *In an oil fired gas turbine plant, air enters at 100kN/m² and 27°C and compressed to 450kN/m². The maximum cycle temperature is 600°C. The isentropic efficiencies of the compressor and turbine are 85% and 86% respectively. The calorific value of oil used is 42,000 kJ/kg. The air flow rate is 2 kg/s. Calculate*

(i) air-fuel ratio,

(ii) Power developed by plant and

(iii) thermal efficiency of the plant.

Take, $C_P = 1.005\ kJ/kgK$ *and* $\gamma = 1.4$ *for air and* $C_P = 1.11\ kJ/kgK$ *and* $\gamma = 1.33$ *for gases.*

Solution:- Given data: Refer Fig 9.8,

$p_1 = 100\ kN/m^2, p_2 = 450\ kN/m^2, T_1 = 27°C = 300\ K, T_3 = 600°C = 873\ K, \eta_c = 0.85$

$\eta_t = 0.86,\ CV = 42,000\ kJ/kg, m_a = 2\ kg/s,\ A/F = ?,\ P = ?, \eta_{th} = ?$

For process 1-2

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \qquad \therefore T_2 = 300 \times \left(\frac{450}{100}\right)^{\frac{1.4-1}{1.4}}$$

$\therefore$ Ideal temperature of air after compression $T_2 = 461.05K$

Now, isentropic efficiency of compressor

$$\eta_c = \frac{T_2 - T_1}{T_2' - T_1}$$

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∴ Actual temperature of air after compression

$$\therefore T_2' = T_1 + \frac{T_2 - T_1}{\eta_c} \qquad = 300 + \frac{461.05 - 300}{0.85}$$

$$\therefore T_2' = 489.47K$$

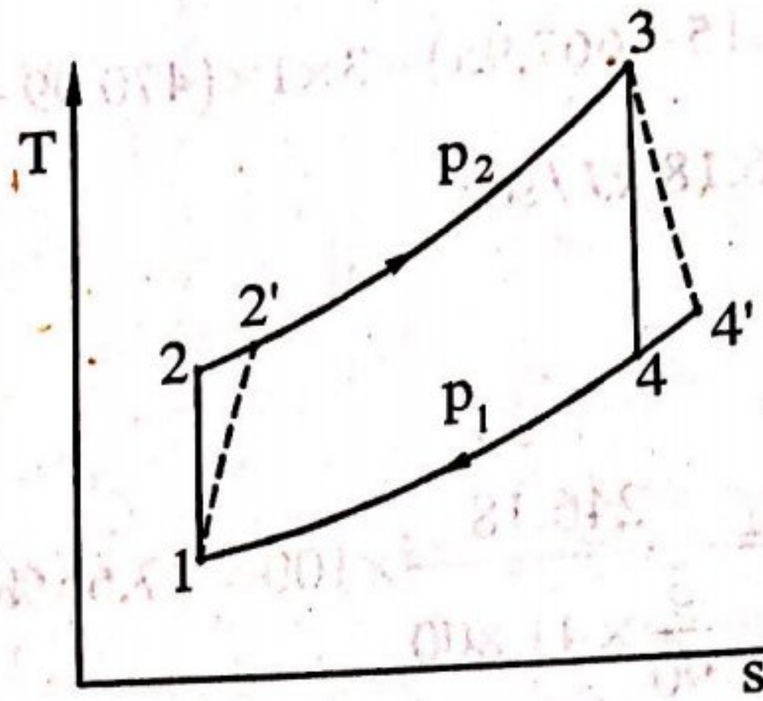


Fig. 9.8

For process 3-4

$$\frac{T_3}{T_4} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \qquad \therefore T_4 = 873 \times \left(\frac{100}{450}\right)^{\frac{1.33-1}{1.33}}$$

∴ Ideal temperature of gases after expansion in turbine $T_4 = 601.09K$

Now, isentropic efficiency of turbine

$$\eta_t = \frac{T_3 - T_4'}{T_3 - T_4}$$

∴ Actual temperature of gases after expansion in turbine

$$T_4' = T_3 - \eta_t (T_3 - T_4) \quad = 873 - 0.86(873 - 601.09) = 639.15\,K$$

Now, Mass of hot gases = Mass of air + Mass of fuel supplied

$$\therefore m_g = m_a + m_f$$

Heat balance for combustion chamber :

Heat supplied to combustion chamber = Heat gain by working fluid

$$\therefore m_f \times CV \times \eta_{comb} = m_g C_{pg} T_3 - m_a C_{pa} T_2'$$

Assuming η_{comb} = Combustion efficiency = 100%

$$\therefore m_f \; CV \times 1 = m_g C_{pg} T_3 - m_a C_{pa} T_2'$$

$$\therefore m_f \times 42,000 = (m_a + m_f) \times 1.11 \times 873 - m_a \times 1.005 \times 489.47$$

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$$\therefore 42000 = \left(\frac{m_a}{m_f} + 1\right) \times 969.03 - \frac{m_a}{m_f} \times 491.91$$

$$\therefore 42000 = \frac{m_a}{m_f} \times 477.11 + 969.03$$

$$\therefore 41030.97 = \frac{m_a}{m_f} \times 477.11 \qquad \therefore \frac{m_a}{m_f} = 85.99$$

$\therefore$ Air fuel ratio, A/F = **85.99 : 1** ***Ans***

Compressor work $W_c = m_a C_{pa}(T_2' - T_1)$

$$= 2 \times 1.005 \times (489.47 - 300) = 380.83 \ kJ/s$$

Turbine work $W_T = m_g \times C_{pg} \times (T_3 - T_4')$

$$= (m_a + m_f) \times C_{pg}(T_3 - T_4')$$

$$= m_a \left(1 + \frac{m_f}{m_a}\right) \times C_{pg}(T_3 - T_4')$$

$$= 2 \times \left(1 + \frac{1}{85.99}\right) \times 1.11 \times (873 - 639.15)$$

$$= 525.18 \ kJ/s$$

Net work produced $W_{net} = W_T - W_C = 525.18 - 380.83 = 144.35 \ kJ/s$

Power developed by plant = **144.35 kW** ***Ans***

Thermal efficiency $\eta_{th} = \dfrac{W_{net}}{Q_s}$

But $Q_s = m_g C_{pg} T_3 - m_a C_{pa} T_2' = m_a \left[\left(1 + \dfrac{m_f}{m_a}\right) C_{pg} T_3 - C_{pa} T_2'\right]$

$$= 2 \times \left[\left(1 + \frac{1}{85.99}\right) \times 1.11 \times 873 - 1.005 \times 489.47\right] = 976.76 \ kJ/s$$

$$\therefore \eta_{th} = \frac{144.35}{976.76} \times 100 = \mathbf{14.77\ \%} \qquad \textit{Ans}$$

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Problem 9.4 *In a gas turbine power plant, the air is taken in at 1 bar and 27°C and is compressed to 5 bar with an adiabatic efficiency of 84%. Heat is added in the combustion chamber to raise the temperature 620°C. The expansion efficiency of turbine is 86%. Calculate for 1 kg/s air flow : (i) Final temperature of exhaust gases, (ii) Air-fuel ratio, (iii) Net power output, and (iv) Overall thermal efficiency.*

Take, $C_p = 1$ *kJ/kg K,* $C_v = 0.714$ *kJ/kg K for air,*

$C_p = 1.12$ *kJ/kg K,* $C_v = 0.842$ *kJ/kg K for gases*

Calorific value of fuel used is 42,000 kJ/kg.

Solution : Given data : $p_1 = 1$ bar, $T_1 = 27°C = 300$ K, $p_2 = 5$ bar, $\eta_c = 0.84$, $T_3 = 620°C = 893$ K, $\eta_t = 0.86$, $m_a = 1$ kg/s, $T_4' = ?$, A/F $= ?$, $W_{net} = ?$, $\eta_{th} = ?$

For air $\gamma = \dfrac{C_p}{C_v} = \dfrac{1}{0.714} = 1.4$, For gases $\gamma = \dfrac{C_p}{C_v} = \dfrac{1.12}{0.842} = 1.33$

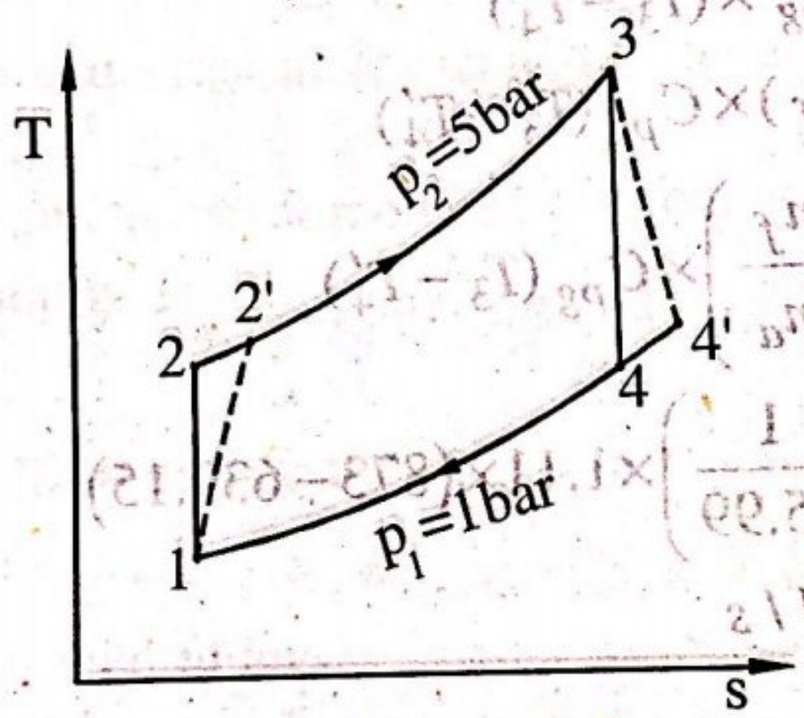


Fig. 9.9

For isentropic process 1-2 (Refer Fig. 6.10)

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \qquad \therefore T_2 = 300 \times \left(\frac{5}{1}\right)^{\frac{1.4-1}{1.4}} = 475.14\,K$$

$$\text{Also, } \eta_c = 0.84 = \frac{T_2 - T_1}{T_2' - T_1} \qquad \therefore T_2' = 300 + \frac{475.14 - 300}{0.84} = 508.51\,K$$

For isentropic process 3-4

$$\frac{T_3}{T_4} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \qquad \therefore T_4 = 893 \times \left(\frac{1}{5}\right)^{\frac{1.33-1}{1.33}} = 598.99\,K$$

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Also, $\eta_t = 0.86 = \dfrac{T_3 - T_4'}{T_3 - T_4}$ $\quad \therefore T_4' = 893 - 0.86(893 - 598.99) = 640.15\ K$

(i) Final temperature of exhaust gases $\therefore T_4' = 640.15\ K =$ **367.15°C** ***Ans***

Now, heat added to the combustion chamber

$$m_f \cdot CV = (m_a + m_f) \cdot C_{pg} T_3 - m_a C_{pa} T_2'$$

$$\therefore CV = \left(\frac{m_a}{m_f} + 1\right) \cdot C_{pg} \cdot T_3 - \frac{m_a}{m_f} \cdot C_{pa} \cdot T_2'$$

$$\therefore 42{,}000 = \left(\frac{m_a}{m_f} + 1\right) \times 1.12 \times 893 - \frac{m_a}{m_f} \times 1 \times 508.51$$

$$\therefore 42{,}000 = 491.65 \times \frac{m_a}{m_f} + 1000.16$$

$$\therefore \frac{m_a}{m_f} = 83.39$$

(ii) Air fuel ratio A/F = **83.39 : 1** ***Ans***

Now, compressor work $W_C = m_a \cdot C_{pa} (T_2' - T_1)$

$$= 1 \times 1 \times (508.51 - 300)$$

$$= 208.51 \text{ kJ/s}$$

Turbine work $W_T = (m_a + m_f) \cdot C_{pg} \cdot (T_3 - T_4')$

$$= \left(1 + \frac{1}{83.39}\right) \times 1.12 \times (893 - 640.15)$$

$$= 286.58 \text{ kJ/s}$$

(iii) Net work produced $W_{net} = W_T - W_C = 286.58 - 208.51 =$ **78.078 kW** ***Ans***

(iv) Overall thermal efficiency

$$\eta_{th} = \frac{W_{net}}{Q_s} = \frac{W_{net}}{m_f \times CV} = \frac{78.078}{\frac{1}{83.39} \times 42{,}000} \times 100 = \mathbf{15.5\ \%} \qquad \textit{Ans}$$

Problem 9.5 *Air enters the compressor of an open cycle, constant pressure gas turbine at 1 bar and 27°C. The pressure of air after compression becomes 4 bar. The isentropic*

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efficiencies of compressor and turbine are 78% and 84% respectively. The air fuel ratio used is 75 : 1. If the rate of flow of air is 2.5 kg/sec, determine the power developed and thermal efficiency of the cycle. Take $C_p = 1.005$ kJ/kg K and $\gamma = 1.4$ both for air and gases. The calorific value of fuel used = 42,000 kJ/kg. **[Same as May '12]**

Solution : Given data : $p_1 = 1$ bar, $T_1 = 27°C = 300$ K, $p_2 = 4$ bar, $\eta_c = 78\%$, $\eta_t = 84\%$, A/F = 75 : 1, $m_a = 2.5$ kg/sec, P = ?, η_{th} = ?, $C_p = 1.005$ kJ/kg K, $\gamma = 1.4$, C.V. = 42,000 kJ/kg

For isentropic process 1-2 (Refer Fig. 9.10)

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \qquad \therefore T_2 = 300 \times \left(\frac{4}{1}\right)^{\frac{1.4-1}{1.4}} = 445.79\ K$$

Also, $\eta_c = 0.78 = \dfrac{T_2 - T_1}{T_2' - T_1} \qquad \therefore\ T_2' = 300 + \dfrac{445.79 - 300}{0.78} = 486.92$ K

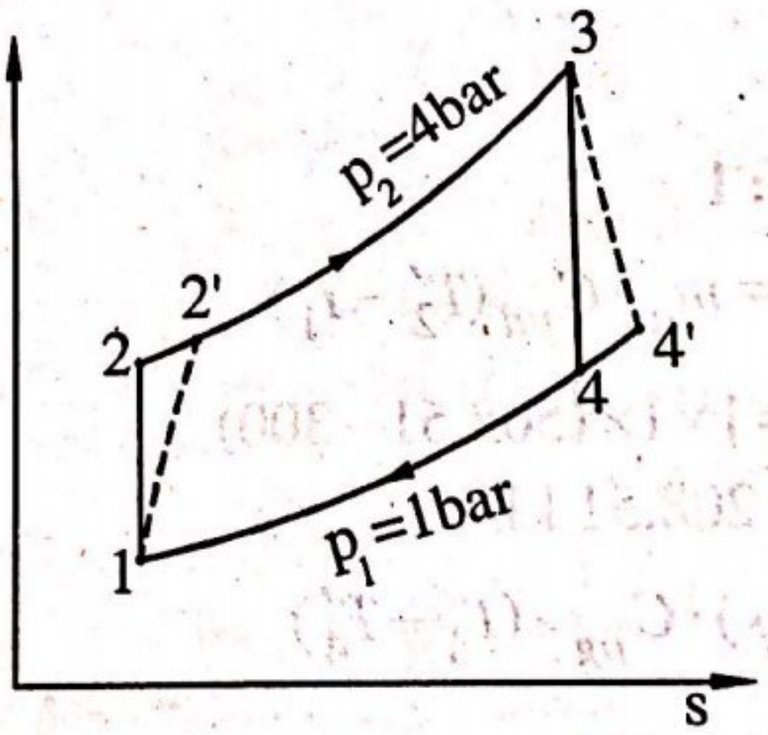


Fig. 9.10

Heat balance for combustion chamber

$$m_f \cdot CV = (m_a + m_f) \cdot C_p \cdot T_3 - m_a C_p T_2'$$

$$\therefore CV = \left(\frac{m_a}{m_f} + 1\right) \cdot C_p \cdot T_3 - \frac{m_a}{m_f} \cdot C_p \cdot T_2'$$

$$\therefore 42,000 = (75+1) \times 1.005 \times T_3 - 75 \times 1.005 \times 486.92$$

$$\therefore T_3 = 1030.39 \text{ K}$$

For isentropic process 3-4

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$$\frac{T_3}{T_4} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \qquad T_4 = 1030.39 \times \left(\frac{1}{4}\right)^{\frac{1.4-1}{1.4}} = 693.4\ K$$

$$\text{Also, } \eta_t = 0.84 = \frac{T_3 - T_4'}{T_3 - T_4}$$

$$\therefore\ T_4 = 1030.39 - 0.84(1030.39 - 693.4) = 747.31\,K$$

Net work produced by plant

$$W_{net} = W_T - W_C$$

$$= (m_a + m_f) \cdot C_p \cdot (T_3 - T_4') - m_a \cdot C_p \cdot (T_2' - T_1)$$

$$= m_a \left[\left(1 + \frac{m_f}{m_a}\right) C_p \cdot (T_3 - T_4') - C_p \cdot (T_2' - T_1)\right]$$

$$= m_a \cdot \left[\left(1 + \frac{1}{75}\right) \times 1.005 \times (1030.39 - 747.31) - 1.005 \times (486.92 - 300)\right]$$

$$= 100.43\ m_a \text{ kJ/s}$$

$$= 100.43 \text{ kJ/kg of air}$$

Power developed by plant $= W_{net} \times m_a$

$$= 100.43 \times 2.5 = \mathbf{251.08\ kW} \qquad \textit{Ans}$$

Now, heat supplied to the combustion chamber

$$Q_s = m_f \times CV = \left(\frac{m_f}{m_a} \cdot CV\right) \times m_a = \left(\frac{1}{15} \times 42{,}000\right) \times m_a = 560\, m_a\ kJ/s$$

$$= 560 \text{ kJ/kg of air}$$

Thermal efficiency of cycle

$$\eta_{th} = \frac{W_{net}}{Q_s} = \frac{100.08\, m_a}{560\, m_a} \times 100 = \mathbf{17.93\ \%} \qquad \textit{Ans}$$

Problem 9.6 *A gas turbine unit has a pressure ratio of 6 and maximum cycle temperature is 900°C. The isentropic efficiencies of the compressor and turbine are 85% and 90% respectively. Air enters the compressor at 15°C at the rate of 5 kg/s. Calorific value of the fuel used is 43,000 kJ/kg, combustion efficiency is 95%. Using* C_{pa} *= 1 kJ/kg K,*

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C_{pg} = 1.07 kJ/kg K and γ = 1.4 for air and gases, find (i) thermal efficiency, (ii) power output, (iii) air-fuel ratio and (iv) specific fuel consumption. **[May '15]**

Solution : Given data : $p_2 / p_1 = 6$, $T_3 = 900°C = 1173$ K, $\eta_c = 0.85$, $\eta_t = 0.9$, $T_1 = 15^{\circ} C = 288K$, $m_a = 5$ kg/s, C.V. = 43,000 kJ/kg, $\eta_{comb} = 0.95$, $C_{pa} = 1$ kJ/kg K, $C_{pg} = 1.07$ kJ/kg K, $\gamma = 1.4$, $\eta_{th} = ?$, $P = ?$, $A/F = ?$, $s_{fc} = ?$

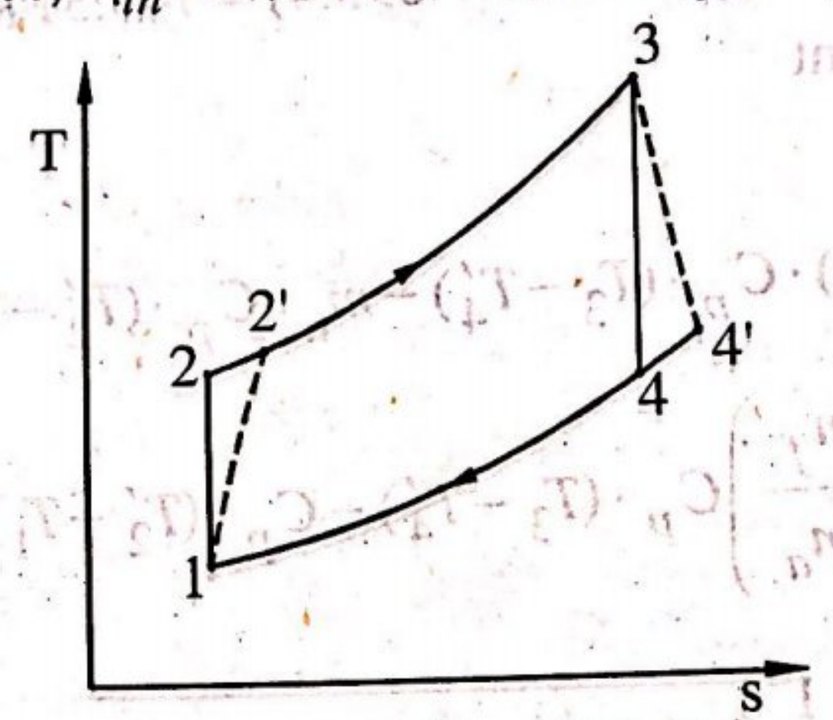


Fig. 9.11

For isentropic compression process (Refer Fig. 9.11)

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \qquad \therefore T_2 = 288 \times (6)^{\frac{1.4-1}{1.4}} = 480.53\,K$$

$$\text{Also, } \eta_c = 0.85 = \frac{T_2 - T_1}{T_2' - T_1} \qquad \therefore T_2' = 288 + \frac{480.53 - 288}{0.85} = 514.5\,K$$

For isentropic expansion process

$$\frac{T_3}{T_4} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \qquad \therefore T_4 = T_3\left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = 1173 \times \left(\frac{1}{6}\right)^{\frac{1.4-1}{1.4}} = 703.02\,K$$

$$\text{Also, } \eta_t = 0.9 = \frac{T_3 - T_4'}{T_3 - T_4} \qquad \therefore T_4' = 1173 - 0.9(1173 - 703.02) = 750.02\,K$$

Energy balance for combustion chamber

$$m_f \times CV \times \eta_{comb} = (m_a + m_f) C_{pg} T_3 - m_a C_{pa} T_2'$$

$$\therefore m_f \times 43,000 \times 0.95 = (5 + m_f) \times 1.07 \times 1173 - 5 \times 1 \times 514.5$$

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$$\therefore\ 40850 m_f = 1255.11 m_f + 3703.05$$

$$\therefore\ m_f = 0.093 \text{ kg/s}$$

$$\text{Air-fuel ratio} = \frac{m_a}{m_f} = \frac{5}{0.093} = \mathbf{53.46 : 1} \qquad \textit{Ans}$$

Net power output

$$P = W_T - W_C$$

$$= (m_a + m_f) C_{pg} (T_3 - T_4') - m_a C_{pa} (T_2' - T_1)$$

$$= (5 + 0.093) \times 1.07 \times (1173 - 750.02) - 5 \times 1 \times (514.5 - 288)$$

$$P = \mathbf{1172.53\ kW} \qquad \textit{Ans}$$

Now, heat supplied to the combustion chamber

$$Q_s = m_f \times CV = 0.093 \times 43,000 = 3999\ kW$$

Thermal efficiency of the plant

$$\eta_{th} = \frac{P}{Q_s} = \frac{1172.53}{3999} \times 100 = \mathbf{29.32\%} \qquad \textit{Ans}$$

$$\text{Specific fuel consumption} = \frac{\text{Fuel supplied in kg/hr}}{\text{Power output}}$$

$$= \frac{0.093 \times 3600}{1172.53} = \mathbf{0.2855\ kg/kWh} \qquad \textit{Ans}$$

Problem 9.7 *A simple open cycle gas turbine takes air at 1 bar and 18^{0}C and compresses it upto 5 bar. The pressure drop in combustion chamber is 0.06 bar. The gases enters the turbine at 550^{0}C and expand to 1 bar. The net power developed by plant is 1600 kW. Calculate mass flow of air. Neglect additional mass flow due to fuel. The isentropic efficiencies of compressor and turbine are 85% and 82% respectively. Assume for both air and gases $\gamma = 1.4$ and $C_p = 1.05\ kJ/kgK$.*

Solution: Given data : Refer Fig. 9.12.

$p_1 = 1\ bar$, $T_1 = 18°C = 291\,K$, $p_2 = 5\ bar$. Pressure drop in combustion chamber $\Delta p_2 = 0.06\ bar$, Pressure at inlet of turbine $p_2' = p_2 - \Delta p_2 = 5 - 0.06 = 4.94\ bar$,

$T_3 = 550°C = 823\,K$, $P = 1600\ kW$, $m_a = ?$

$\eta_c = 0.85$, $\eta_t = 0.82$, $\gamma = 1.4$, $C_p = 1.05\ kJ/kgK$

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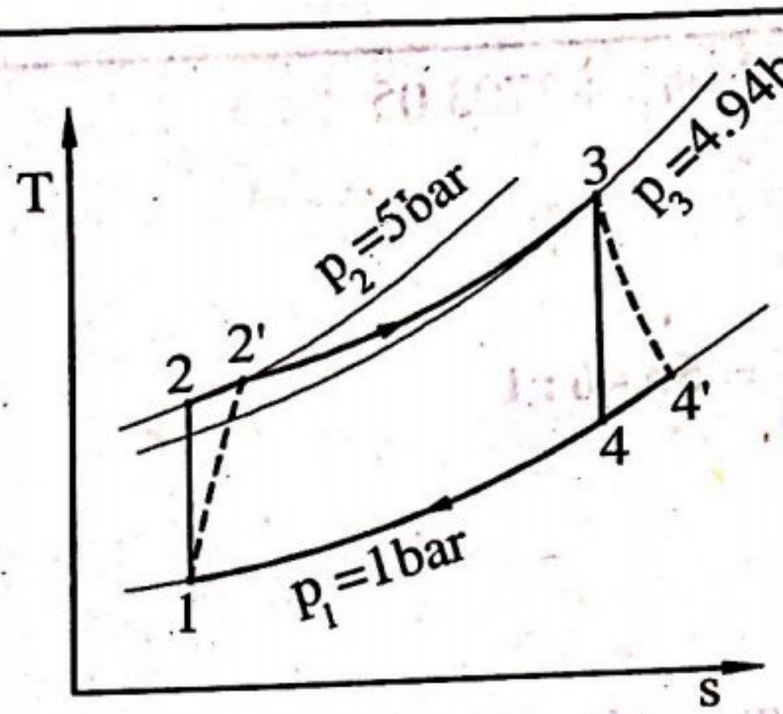


Fig. 9.12

For compression process 1-2:

$$\frac{T_2}{T_1}=\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \quad \therefore T_2=291\times\left(\frac{5}{1}\right)^{\frac{1.4-1}{1.4}}$$

$$\therefore T_2=460.89K$$

But $$\eta_c=0.85=\frac{T_2-T_1}{T_2'-T_1}$$

$$\therefore T_2'=291+\frac{(460.89-291)}{0.85}=490.87K$$

For expansion process 3-4 :

Since, there is a fall of pressure of 0.06 bar through the combnstion chamber hence

the expansion ratio $=\frac{p_2'}{p_1}=\frac{4.94}{1}=4.94$

Now, $$\frac{T_3}{T_4}=\left(\frac{p_2'}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \quad \therefore T_4=823\times\left(\frac{1}{4.94}\right)^{\frac{1.4-1}{1.4}}=521.42K$$

But $\eta_t=0.82=\frac{T_3-T_4'}{T_3-T_4}$ $\quad \therefore T_4'=823-0.82(823-521.42)=575.71K$

Compressor work $W_C=C_p(T_2'-T_1)=1.05\times(490.87-291)=209.86\ kJ/kg$

Turbine work $W_T=C_p(T_3-T_4')=1.05\times(823-575.71)=259.65\ kJ/kg$

Net work $W_{net}=W_T-W_C=259.65-209.86=49.79\ kJ/kg$

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But net power output $P = m_a \times W_{net}$

$$\therefore 1600\ kJ/s = m_a \times 49.79\ kJ/kg$$

$$\therefore m_a = \mathbf{32.13\ kg/s} \qquad \textbf{Ans}$$

Problem 9.8 *In a gas turbine plant, the air at 25°C and 1 bar is compressed to 4 bar with compression efficiency of 80%. The air is heated in the regenerator and the combustion chamber till its temperature is raised to 700°C and during the process the pressure falls by 0.14 bar. The air is then expanded in the turbine and passes to regenerator which has 75% effectiveness, and causes a pressure drop of 0.14 bar. If the isentropic efficiency of the turbine is 85%, determine the thermal efficiency of the plant. Take specific heat of air as 1.005 kJ/kg-K, $\gamma = 1.4$.* **[G.U., Dec. 2002]**

Solution : Given data : $p_1 = 1$ bar, $T_1 = 25°C = 298\ K$, $p_2 = 4$ bar, $\eta_c = 0.8$, $T_3 = 700°C$ $= 973$ K, $p_3 = 4 - 0.14 = 3.86$ bar, $p_4 = 1 + 0.14 = 1.14$ bar, $\varepsilon = 0.75$, $\eta_t = 0.85$, $\eta_{th} = ?$, $C_p = 1.005$ kJ/kg K, $\gamma = 1.4$

For isentropic compression process 1-2 (Refer Fig. 9.12a)

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \qquad \therefore T_2 = 298 \times (4)^{\frac{1.4-1}{1.4}} = 442.82\ K$$

$$\text{Also, } \eta_c = 0.8 = \frac{T_2 - T_1}{T_2' - T_1} \qquad \therefore T_2' = 298 + \frac{442.82 - 289}{0.8} = 479.03\ \text{K}$$

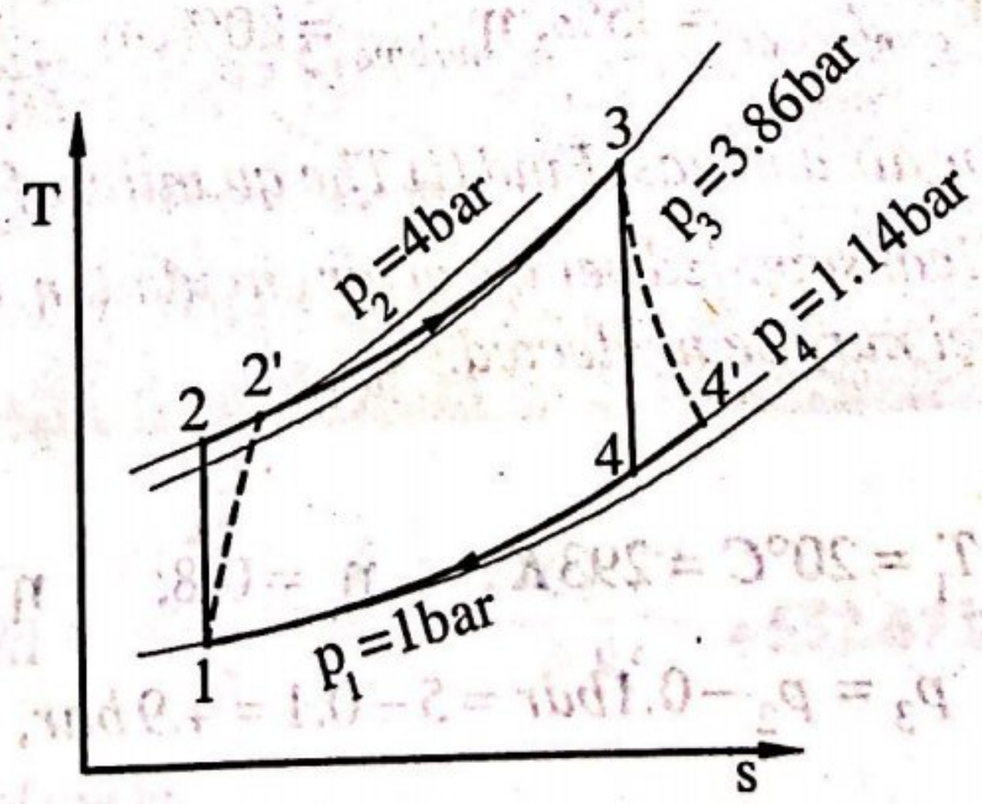


Fig. 9.12a

For isentropic expansion process 3-4

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$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} \qquad \therefore T_4 = T_3 \times \left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} = 973 \times \left(\frac{1.14}{3.86}\right)^{\frac{1.4-1}{1.4}} = 686.71\,K$$

Also, $\eta_t = 0.85 = \dfrac{T_3 - T_4'}{T_3 - T_4}$ $\quad \therefore T_4' = 973 - 0.85(973 - 686.71) = 729.65$ K

Now, effectiveness of heat exchanger (regenerator) is given by

$$\varepsilon = 0.75 = \frac{T_5' - T_2'}{T_4' - T_2'}$$

$$\therefore T_5' = T_2' + 0.75 \times (T_4' - T_2')$$
$$= 479.03 + 0.75 \times (729.65 - 479.03)$$
$$= 666.99 \text{ K}$$

Thermal efficiency of gas turbine with regenerator is given by

$$\eta_{th} = \frac{C_p(T_3 - T_4') - C_p(T_2' - T_1)}{C_p(T_3 - T_5')}$$

$$\therefore \eta_{th} = \frac{(973 - 729.65) - (479.03 - 298)}{(973 - 666.99)} \times 100 = \mathbf{20.36\%} \quad \textit{Ans}$$

Problem 9.9 *In a constant pressure open cycle gas turbine air enters at 1 bar and 20°C and leaves the compressor at 5 bar. Using the following data : Temperature of gases entering the turbine = 680°C, Pressure losses in the combustion chamber = 0.1 bar,* $\eta_{compressor} = 85\%$, $\eta_{turbine} = 80\%$, $\eta_{combustion} = 85\%$, $\gamma = 1.4$ *and* $C_p = 1.024\,kJ/kg\,K$ *for air and gas. Find (i) The quantity of air circulation if the plant develops 1065 kW, (ii) Heat supplied per kg. of air circulation, (iii) The thermal efficiency of the cycle. Mass of fuel may be neglected.* ***[May '13]***

Solution : Given data :

$p_1 = 1\,bar$, $p_2 = 5\,bar$, $T_1 = 20°C = 293\,K$, $\eta_t = 0.8$, $\eta_c = 0.85$, $\eta_{comb.} = 0.85$, $T_3 = 680°C = 953\,K$, $p_3 = p_2 - 0.1\,bar = 5 - 0.1 = 4.9\,bar$, $m_a = ?, Q_s = ?, \eta_{th} = ?$

(Refer Fig. 9.12)

For compression process

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 293\left(\frac{5}{1}\right)^{\frac{1.4-1}{1.4}} = 464.06\,K$$

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Also, $\eta_c = \dfrac{T_2 - T_1}{T_2' - T_1}$

$$\therefore\ T_2' = 293 + \frac{464.06 - 293}{0.85} = 494.24\ K$$

$$\therefore W_c = C_p\left(T_2' - T_1\right)$$

$$= 1.024\ (494.24 - 293) = 206.07\ \text{kJ/kg of air.}$$

For expansion process

$$T_4 = T_3\left(\frac{p_1}{p_3}\right)^{\frac{\gamma-1}{\gamma}} = 953 \times \left(\frac{1}{4.9}\right)^{\frac{1.4-1}{1.4}} = 605.19\ K$$

Also, $\eta_t = \dfrac{T_3 - T_4'}{T_3 - T_4}$,

$$\therefore T_4' = 953 - 0.8(953 - 605.19) = 674.75K$$

$$\therefore W_T = C_p\left(T_3 - T_4'\right) = 1.024(953 - 674.75) = 284.93\ kJ\ /\ kg.of\ air$$

$$W_{net} = W_T - W_C = 284.93 - 206.07 = 78.85\ kJ\ /\ kg\ of\ air$$

(i) Quantity of air circulation

Power, $P = m_a W_{net}$

$$\therefore m_a = \frac{1065}{78.85} = \mathbf{13.5\ kg/s} \qquad \textit{Ans}$$

(ii) Heat supplied per kg. of air circulation

$$Q_s = C_p\left(T_3 - T_2'\right) = 1.024(953 - 294.24)$$

$$= 469.77\ \text{kJ/kg. of air}$$

$$\text{Actual heat supplied} = \frac{C_p\left(T_3 - T_2'\right)}{\eta_{comb}} = \frac{469.77}{0.85} = \mathbf{552.67\ kg/kg\ of\ air} \qquad \textit{Ans}$$

(iii) Thermal efficiency of cycle

$$\eta_{th} = \frac{W_{net}}{Q_s} = \frac{78.85}{552.67} \times 100 = \mathbf{14.26\ \%} \qquad \textit{Ans}$$

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Problem 9.9a *In a closed cycle gas turbine the following data apply :*

Working substance is air, $C_p = 1 kJ/kg\,K$, *and* $\gamma = 1.4$

Ambient temperature	$= 27°C$
Top temperature	$= 823°C$
Pressure at compressor inlet	$= 1\ bar$
Pressure ratio	$= 4$
Compressor efficiency	$= 80\%$
Turbine efficiency	$= 85\%$
Heating value of fuel	$= 41800\ kJ/kg$
Heater loss	$= 10\%$ *of heating value*

Find the following :
(1) Compressor work, (2) Heat supplied, (3) Turbine work, (4) Net work, (5) Thermal efficiency and (6) work ratio. **[Nov. '14]**

Solution : Given data :

$T_1 = 27°C = 300\,K,\ T_3 = 823°C = 1096K,\ p_1 = 1bar,\ r_p = 4,\ \eta_c = 0.8,\ \eta_t = 0.85,$ CV $= 41800$ kJ/kg, $\eta_h = 1 - \frac{10}{100} = 0.9$, $W_c = ?,\ Q_s = ?,\ W_T = ?,\ W_{net} = ?,\ \eta_{th} = ?,$ WR $= ?$

For isentropic compression process

$$\frac{T_2}{T_1} = \left(r_p\right)^{\frac{\gamma-1}{\gamma}} \qquad \therefore T_2 = 300(4)^{\frac{1.4-1}{1.4}} = 445.79\,K$$

But, $\eta_c = 0.8 = \dfrac{T_2 - T_1}{T_2^1 - T_1}$ $\qquad \therefore T_2' = 300 + \dfrac{445.79 - 300}{0.8} = 482.23\,K$

For isentropic expansion process

$$\frac{T_3}{T_4} = \left(r_p\right)^{\frac{\gamma-1}{\gamma}} \qquad \therefore T_4 = 1096\left(\frac{1}{4}\right)^{\frac{1.4-1}{1.4}} = 737.55\,K$$

But, $\eta_t = 0.85 = \dfrac{T_3 - T_4^1}{T_3 - T_4}$ $\qquad \therefore T_4' = 1096 - 0.85(1096 - 737.55) = 791.31K$

Heat balance for heater

$$m_f CV \times \eta_h = m_a C_p T_3 - m_a C_p T_2'$$

$$\therefore \frac{m_a}{m_f} = \frac{CV \cdot \eta_h}{C_p\left(T_3 - T_2'\right)} = \frac{41800 \times 0.9}{1 \times (1096 - 482.23)} = 61.29\,kg/kg\ of\ fuel$$

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$$W_c = m_a C_p \left(T_2' - T_1\right) = 61.21 \times 1 \times (482.23 - 300) = \textbf{11154.29 kJ/kg of fuel} \quad \textit{Ans}$$

$$W_T = m_a C_p \left(T_3 - T_4'\right) = 61.21 \times 1 \times (1096 - 791.31) = \textbf{18650.07 kJ/kg of fuel} \quad \textit{Ans}$$

$$W_{net} = 18650.07 - 11154.29 = \textbf{7495.86 kJ/kg of fuel} \quad \textit{Ans}$$

$$Q_s = m_a C_p \left(T_3 - T_2'\right) = 61.21 \times 1 \times (1096 - 482.23) = \textbf{37568.86 kJ/kg of fuel} \quad \textit{Ans}$$

$$\eta_{th}' = \frac{W_{net}}{Q_s} = \frac{7495.86}{37568.86} \times 100 = \textbf{19.95 \%} \quad \textit{Ans}$$

$$WR = \frac{W_{net}}{W_T} = \frac{7495.86}{18650.07} = \textbf{0.402} \quad \textit{Ans}$$

9.15 Effect of operating variables on thermal efficiency [Sep. '13]

The parameters which affect the performance of the gas turbine plant are compressor input temperature, pressure ratio, turbine inlet temperature, compressor and turbine efficiencies and specific fuel consumption.

(1) Compressor inlet temperature (T_1)

An increase in compressor inlet temperature, the work input to the compressor is increased due to compression process line shifted towards the right side. Hence, thermal

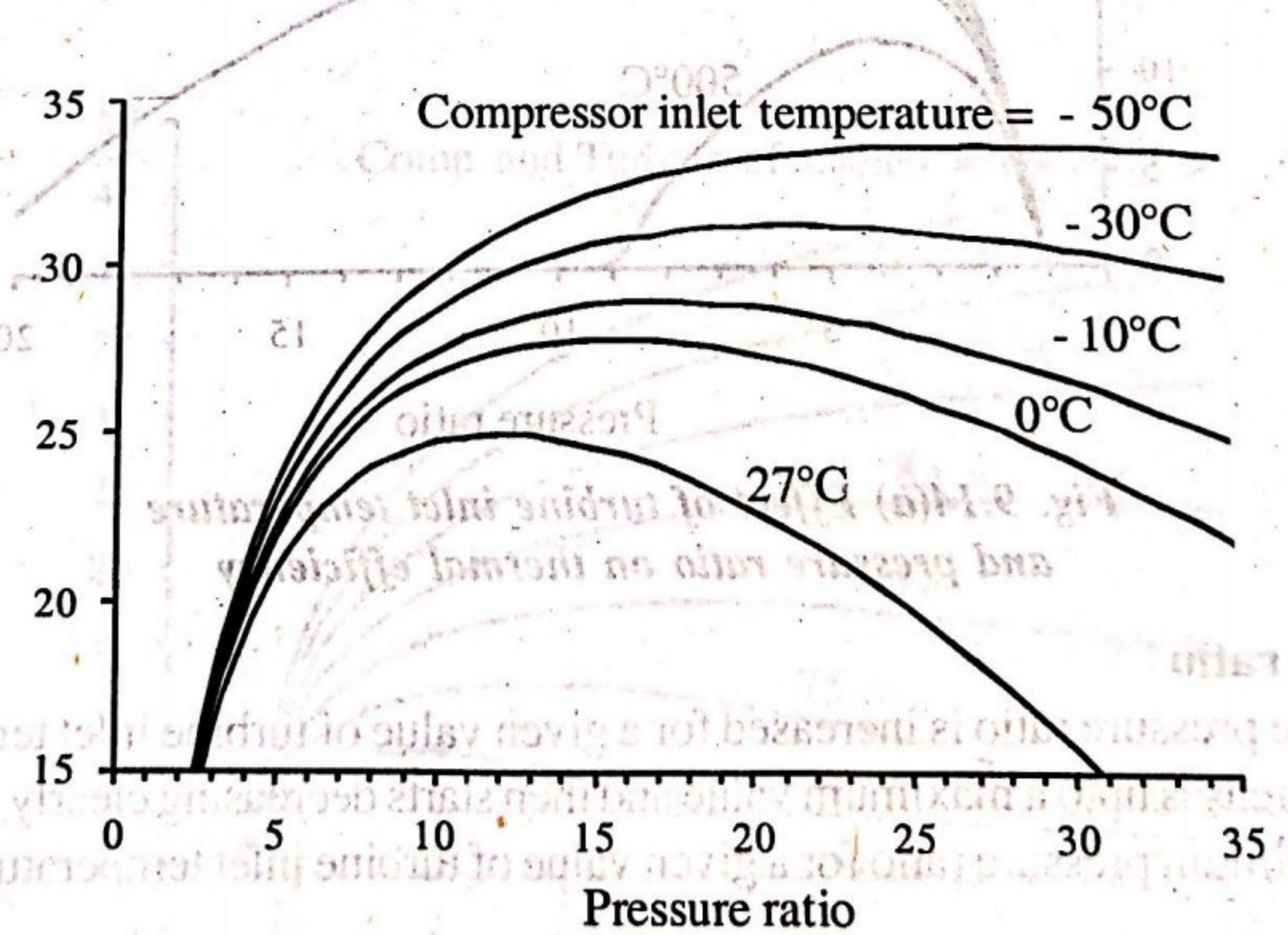


Fig. 9.13 Effect of compressor inlet temperature and pressure ratio on thermal efficiency

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efficiency of the cycle is reduced. However, the heat supplied to the combustion for a given inlet turbine temperature (T_3) is reduced due to increase in temperature of air at outlet of compressor with increase in T_1. The work output of turbine at given T_3 is not affected by T_1. Since rate of increase of compressor work is greater than the rate of reduction in the heat supply, hence net effect of increasing T_1 is a decrease in the thermal efficiency of plant as shown in Fig. 9.13.

(2) Gas turbine inlet temperature (T_3)

An increase in turbine inlet temperature, the work output from the turbine at a given pressure ratio is increased as shown in Fig. 9.14a, but at same time the heat supplied in the combustion chamber is increased. The rate of increase in the turbine work is greater than the rate of increase in heat added and hence, for all pressure ratios increasing the turbine inlet temperature increases the thermal efficiency of cycle. However, the maximum turbine inlet temperature is limited by metallargical considerations. Hence, there is a practical limitation to increasing the turbine inlet temperature is an availability of blading material which can withstand high turbine thermal stresses developed due to high temperature.

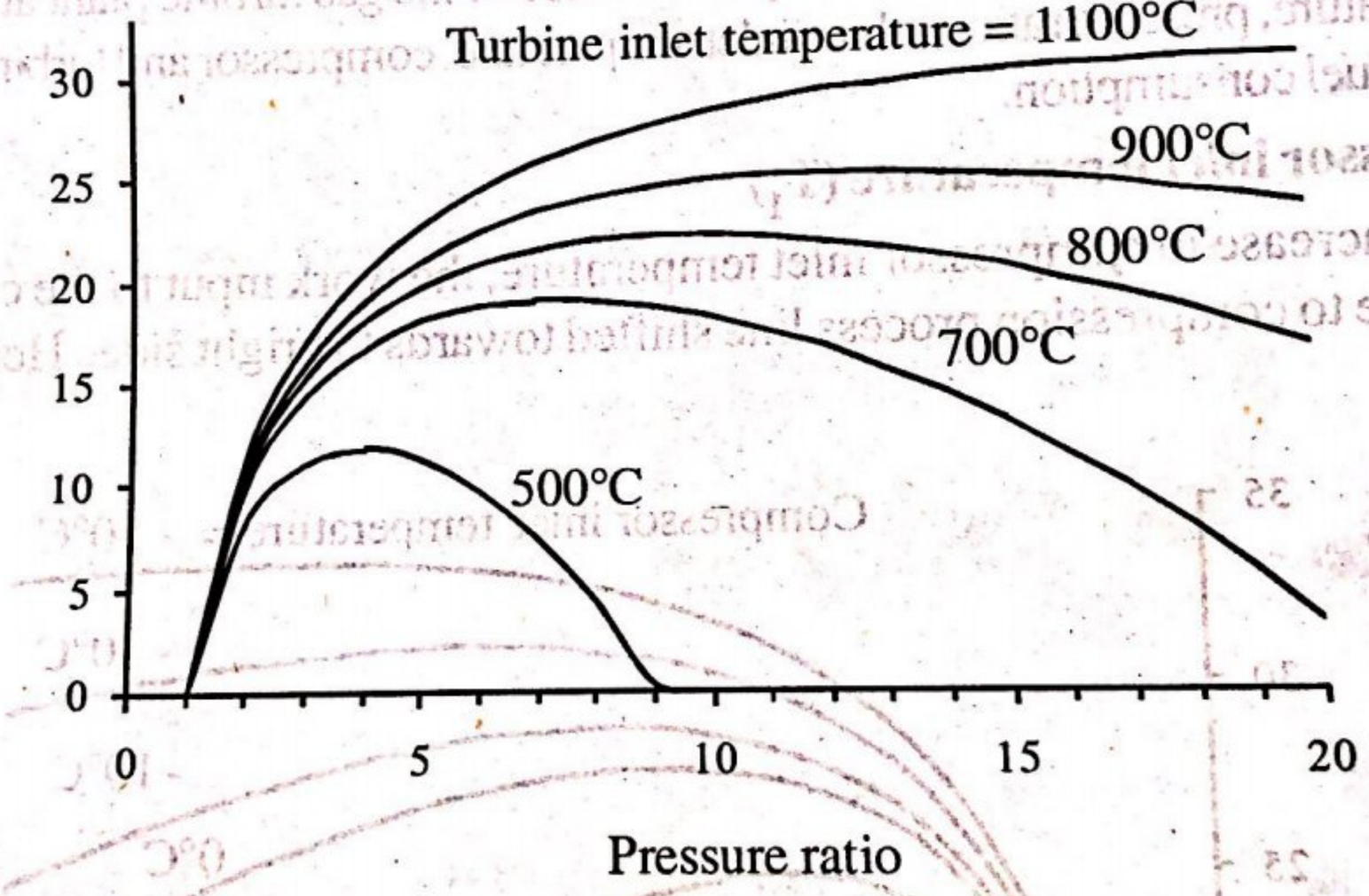


Fig. 9.14(a) Effect of turbine inlet temperature and pressure ratio on thermal efficiency

(3) Pressure ratio

As the pressure ratio is increased for a given value of turbine inlet temperature, the efficiency increases upto a maximum value and then starts decreasing clearly indicating that there is an optimum pressure ratio for a given value of turbine inlet temperature as shown in Fig. 9.14(a).

For a given turbine inlet temperature, as the pressure ratio increases, the heat supplied as well as the heat rejected are reduced as shown in Fig. 9.14(b), hence thermal efficiency is increased. But the rate of change of heat supplied is not the same as the rate of change of heat

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rejected. Therefore, as pressure ratio increases, the thermal efficiency increases until it becomes maximum at optimum pressure ratio and then it drop off with a further increase in pressure ratio. Further, as the turbine inlet temperature increases, the efficiency peak is shifted to higher pressure ratio as shown in Fig. 9.14(a).

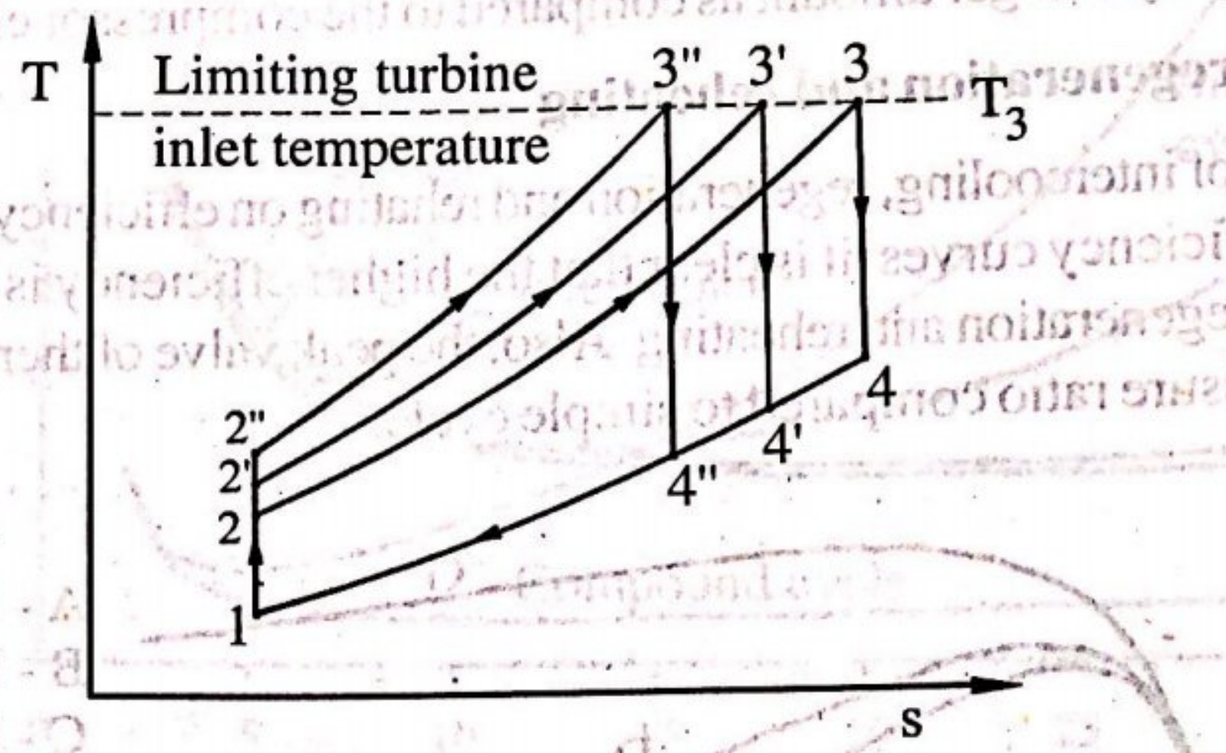


Fig. 9.14(b) Effect of pressure ratio on the Brayton cycle

(4) Compressor and turbine efficiencies

The cycle efficiency greatly depends on the compressor and turbine efficiencies as shown in Fig. 9.15. For a given value of compressor and turbine inlet temperatures the efficiency of a simple cycle is linearly proportional to turbine efficiency. A decrease in compressor efficiency decreases the heat supplied but this decrease in heat supplied is more than offset by the increase in compressor work.

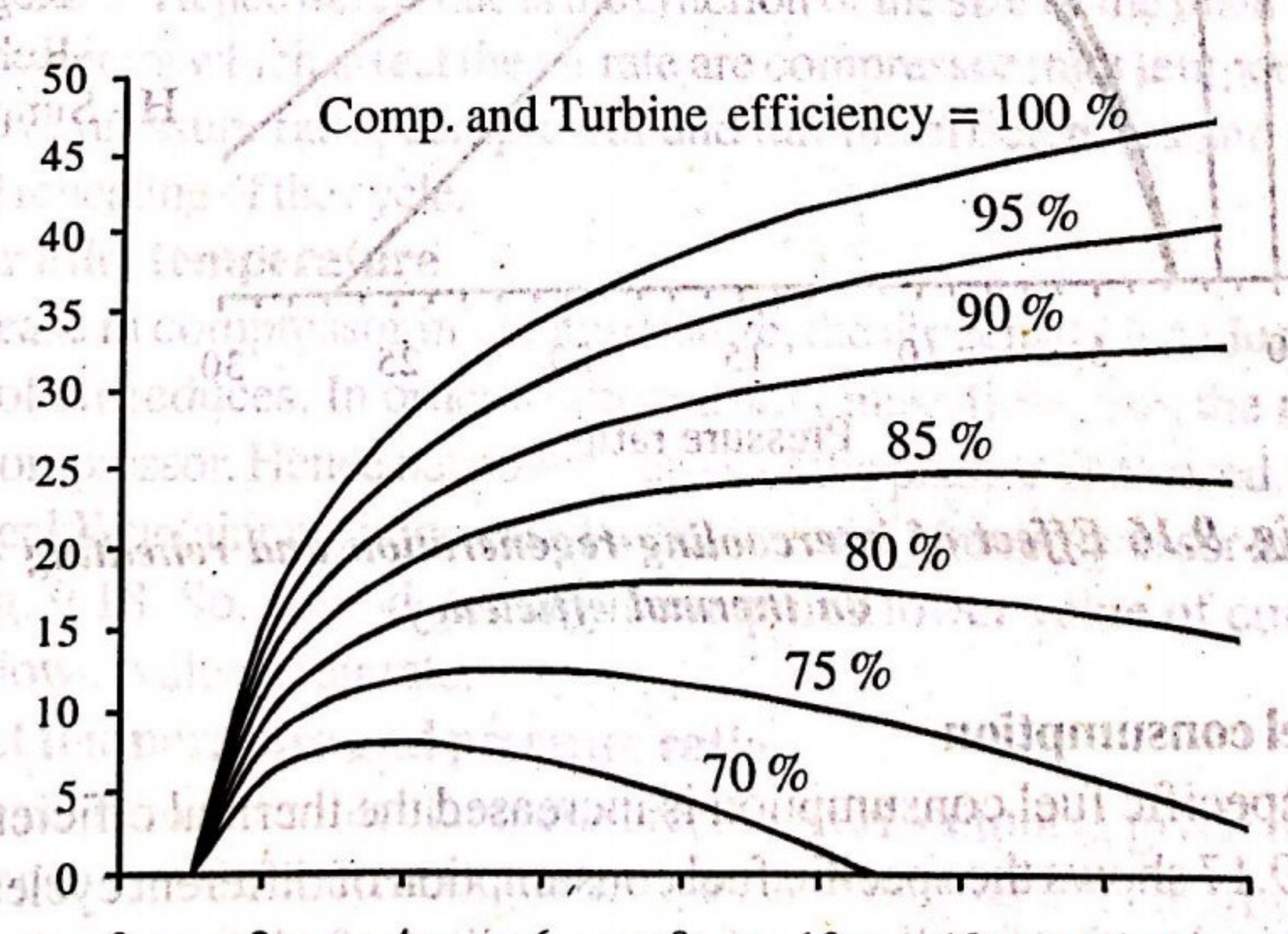


Fig. 9.15 Effect of compressor and turbine efficiencies on thermal efficiency

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From experiment results it is cleat that decrease in compressor and turbine efficiencies of 1%, the cycle efficiency decreased by 3 to 5%. Normally, the turbine has a higher efficiency than the compressor efficiency. Hence turbine develops more power than the compressor consumes for same mass flow of working fluid. Therefore, the loss in turbine efficiency, reduces the cycle efficiency by a larger amount as compared to the compressor efficiency.

(5) Intercooling, regeneration and reheating

The effect of intercooling, regeneration and rehating on efficiency of cycle is shown in Fig. 9.16. From efficiency curves, it is clear that the higher efficiency is possible from cycle with intercooling, regeneration and reheating. Also, the peak valve of thermal efficiency shifts towards lower pressure ratio compared to simple cycle.

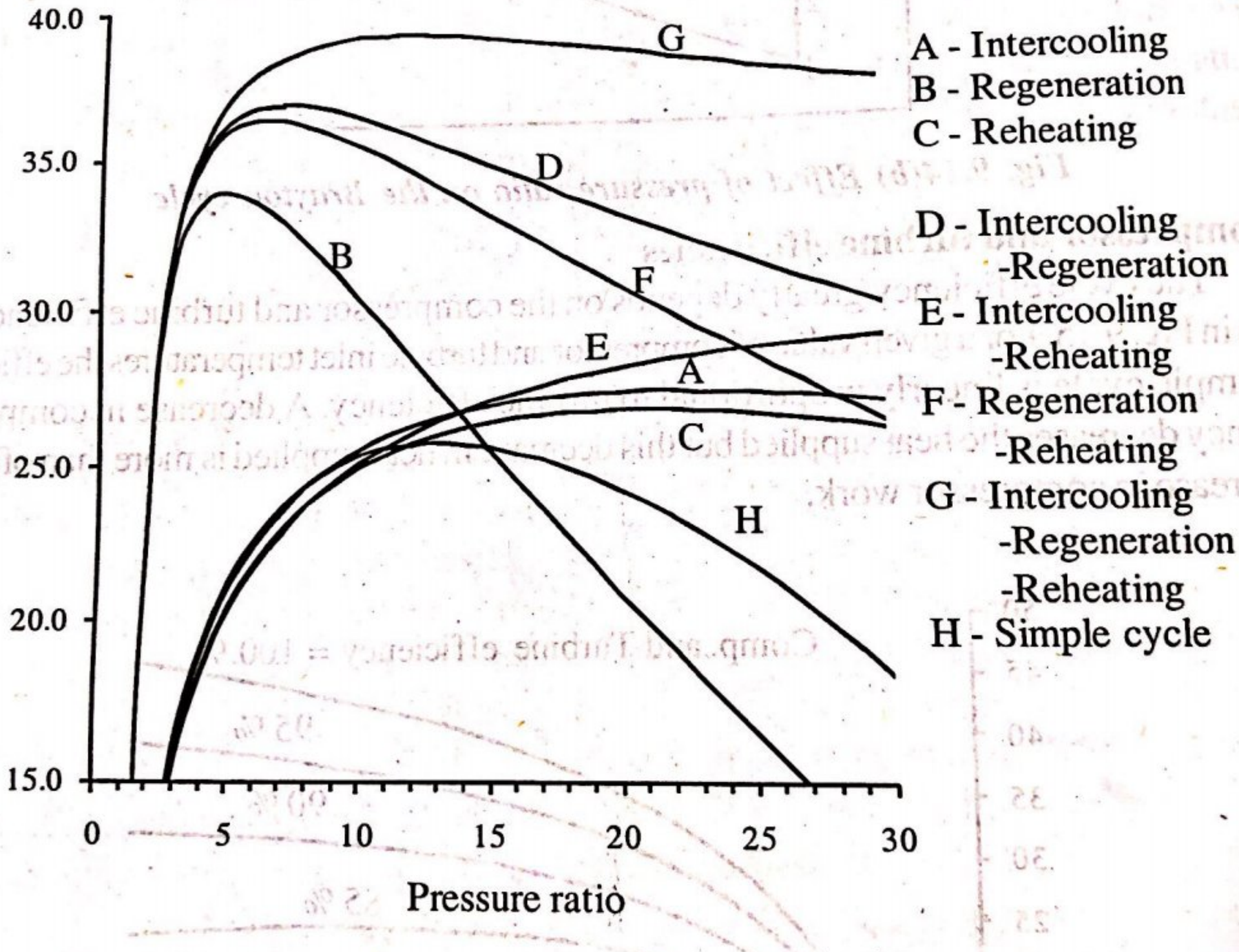


Fig. 9.16 Effect of intercooling regeneration and reheating on on thermal efficiency

(6) Specific fuel consumption

As the specific fuel consumption is increased the thermal efficiency of the cycle decreases. Fig. 9.17 shows the specific fuel consumption of different cycles with respect to pressure ratios. It is clear that for all value of pressure ratio the fuel consumption is the minimum in compound cycle (simple cycle with regeneration, intercooling and reheating).

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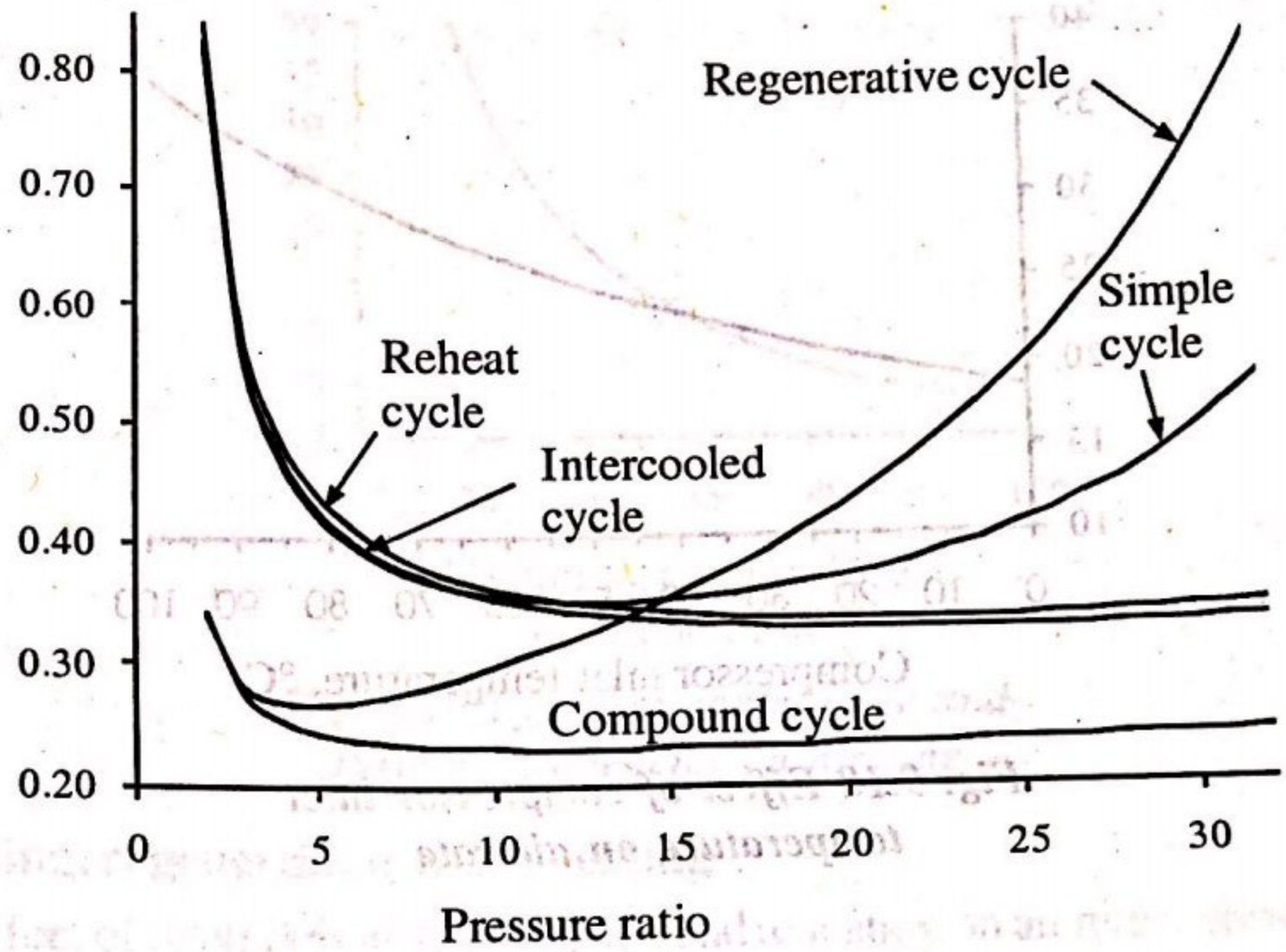


Fig. 9.17 Effect pressure ratio on specific fuel consumption

9.16 Effect of operating variables on air rate

Air rate is defined as the air flow required per kWh output. The thermal efficiency which is measure of fuel economy while the air rate represents the size of the plant. The size of plant is more important when the gas turbine plant is used for aviation. The lower the air rate makes the plant small. Hence the air rate is the criterion of the size of the plant.

The parameters which affect the air rate are compressor inlet temperature, turbine inlet temperature, pressure ratio, compressor and turbine efficiencies and regeneration, intercooling and reheating of the cycle.

(1) Compressor inlet temperature

An increase in compressor inlet temperature, the air density is reduced, and hence mass flow rate of air reduces. In order to get constant mass flow rate, the more power is required to the compressor. Hence net power output of the plant is decreased. In other word, the air required per kWhr (air rate) is increased with increasing the compressor inlet temperature as shown in Fig. 9.18. So, a good gas turbine requires lower value of compressor inlet temperature i.e. lower value of air rate.

(2) Turbine inlet temperature and pressure ratio

An increase in turbine inlet temperature, the work output at given pressure ratio is increased and hence air rate decreases with increase in turbine inlet temperature at given pressure ratio as shown in Fig. 9.19. It is also clear from the figure that as pressure ratio at given temperature increases the air rate decreases until it becomes minimum at optimum pressure ratio and then air rate increases with further increase in pressure ratio. This is due to increase in power output upto certain pressure ratio and than it reduces.

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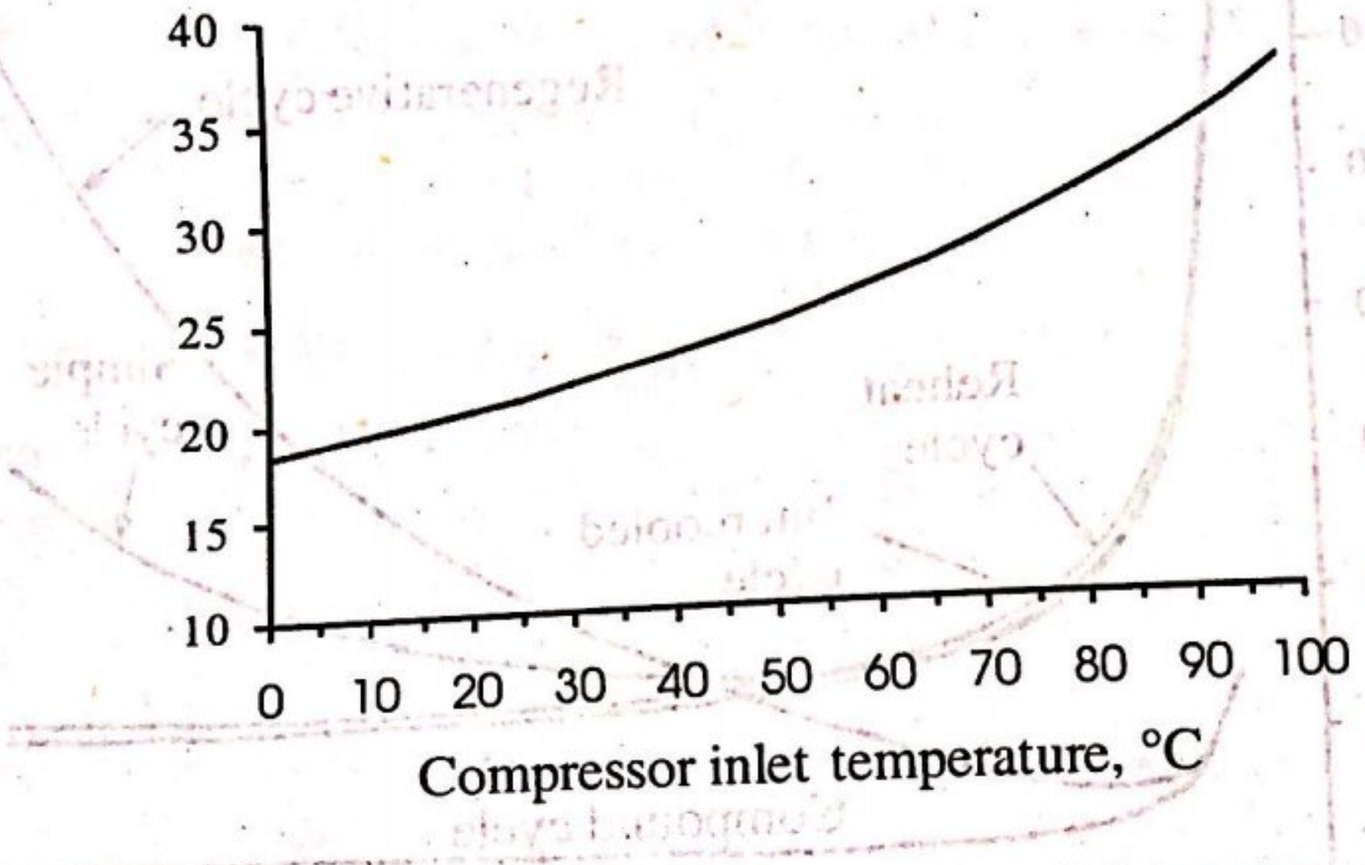


Fig. 9.18 Effect of compressor inlet temperature on air rate

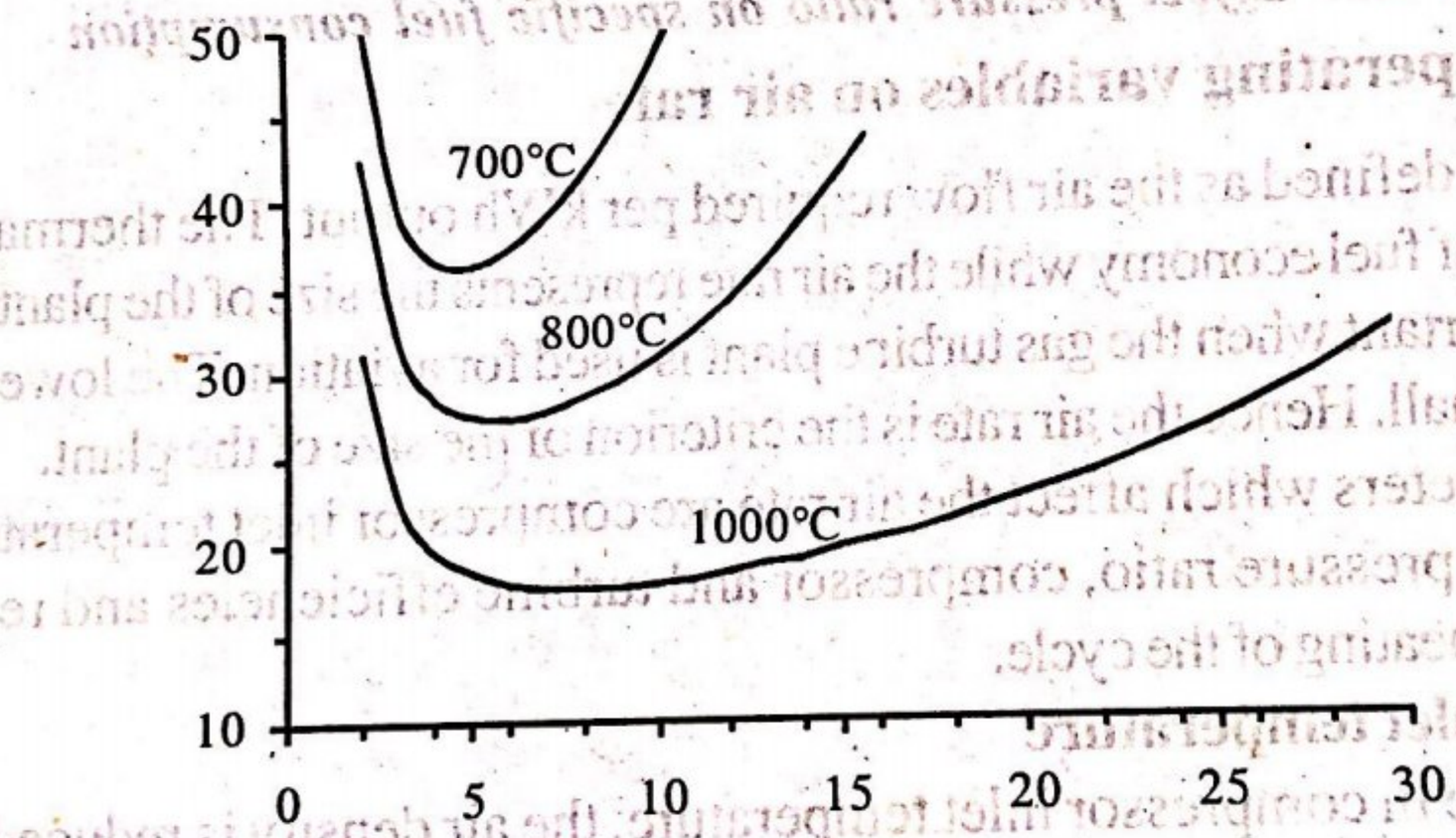


Fig. 9.19 Effect of turbine inlet temperature and pressure ratio on air rate

(3) Compressor and turbine efficiencies

As the efficiencies of the compressor and turbine increases, the net work output incrreases at same mass of air flow. Hence mass flow rate of air per kW is reduced. It means that the air flow rate decreases with increase in efficiency of compressor and turbine as shown in Fig. 9.20.

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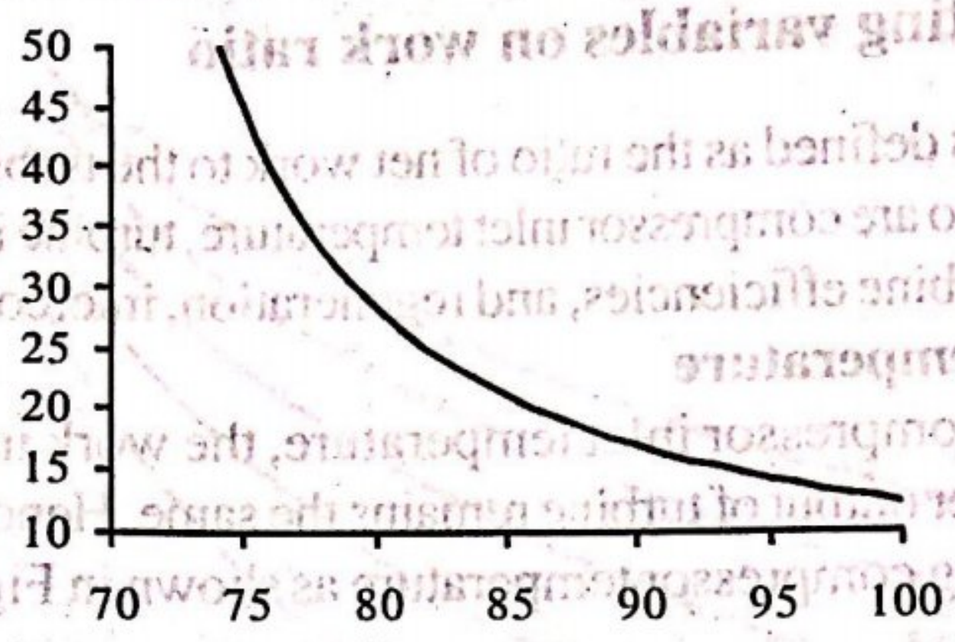


Fig. 9.20 Effect of compressor and turbine efficiencies on air rate

(4) Intercooling, regeneration and reheating

The effect of intercooling, regeneration and reheating on air rate is shown in Fig. 9.21. From figure, it is clear that the air rate is lower in case of cycle with intercooling and reheating, and compound cycle (intercooling, regeneration and reheating) compared to the simple cycle.

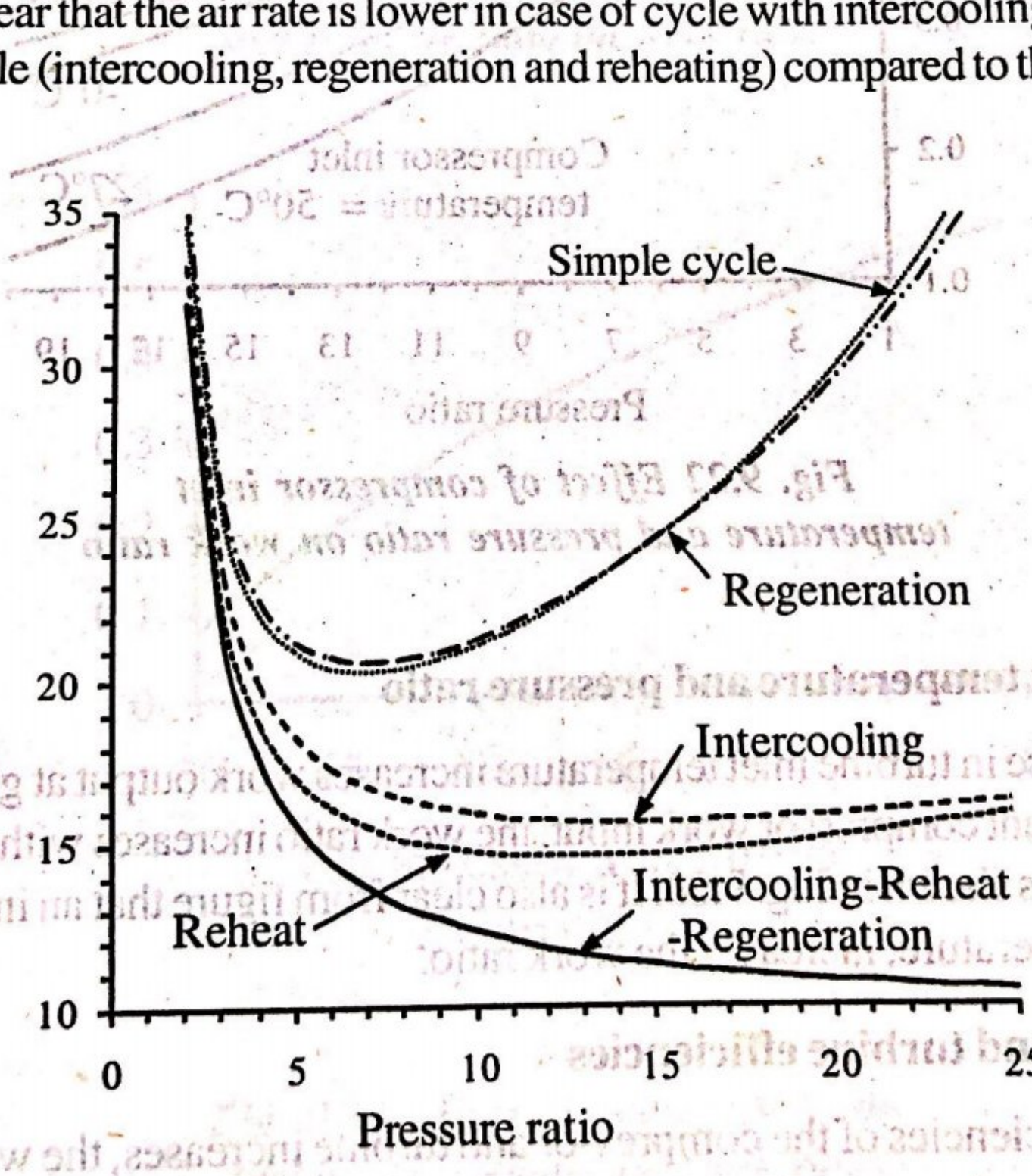


Fig. 9.21 Effect of intercooling, regeneration and reheating on air rate

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9.17 Effect of operating variables on work ratio

The work ratio is defined as the ratio of net work to the turbine work. The parameters which affect the work ratio are compressor inlet temperature, turbine inlet temperature, pressure ratio, compressor and turbine efficiencies, and regeneration, intercooling and reheating.

(1) Compressor inlet temperature

An increase in compressor inlet temperature, the work input to the compressor is increased while the power output of turbine remains the same. Hence net work output of plant is reduced with increasing compressor temperature as shown in Fig. 9.22.

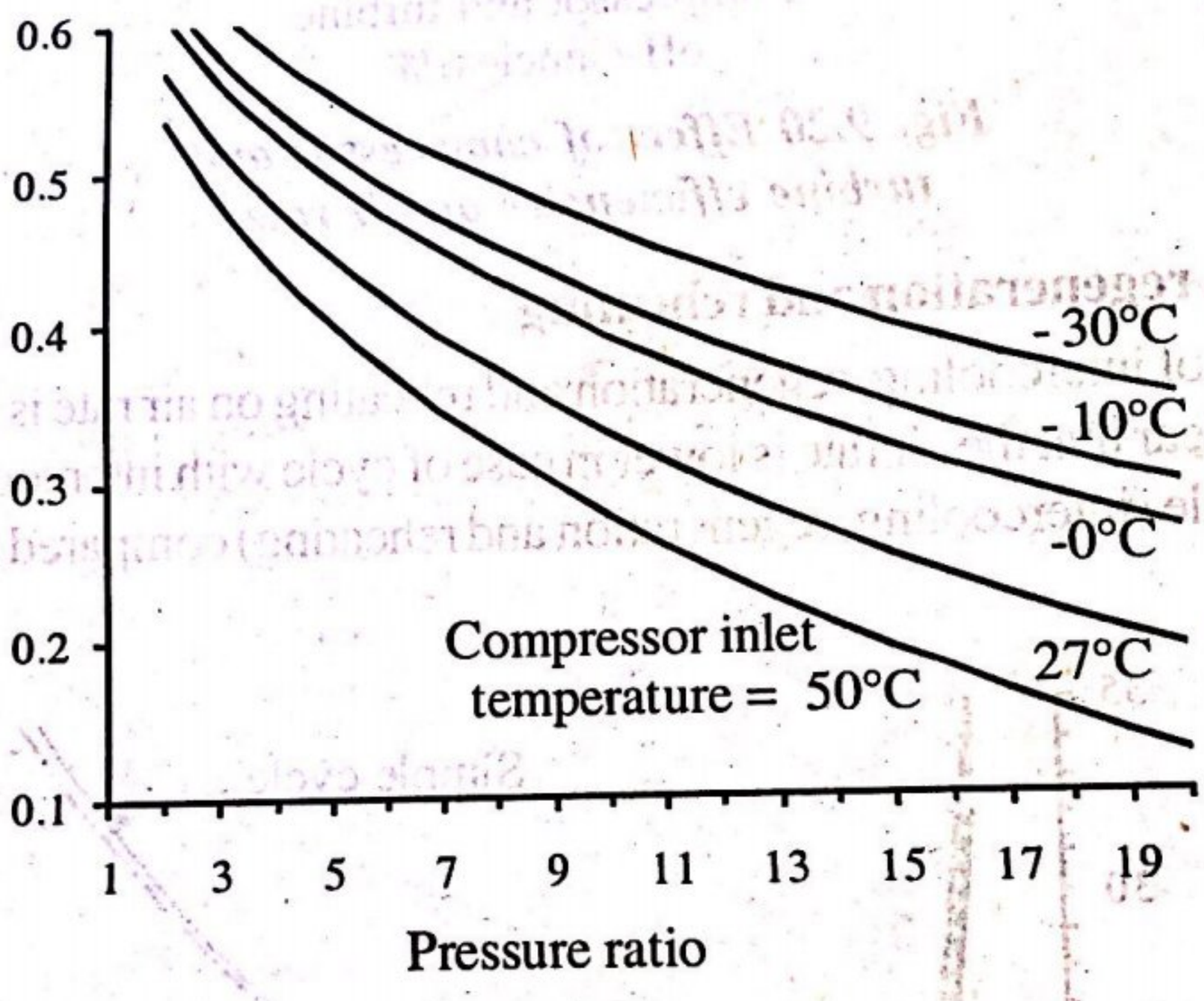


Fig. 9.22 Effect of compressor inlet temperature and pressure ratio on work ratio

(2) Turbine inlet temperature and pressure ratio

An increase in turbine inlet temperature increases work output at given pressure ratio and hence at constant compressor work input, the work ratio increases with increasing turbine inlet temperature as shown in Fig. 9.23. It is also clear from figure that an increase in pressure ratio at given temperature, increases the work ratio.

(3) Compressor and turbine efficiencies

As the efficiencies of the compressor and turbine increases, the work ratio increases due to increase in work output of turbine and reduction in compressor work input. The effect of compressor and turbine efficiencies on work ratio is shown in Fig. 9.24.

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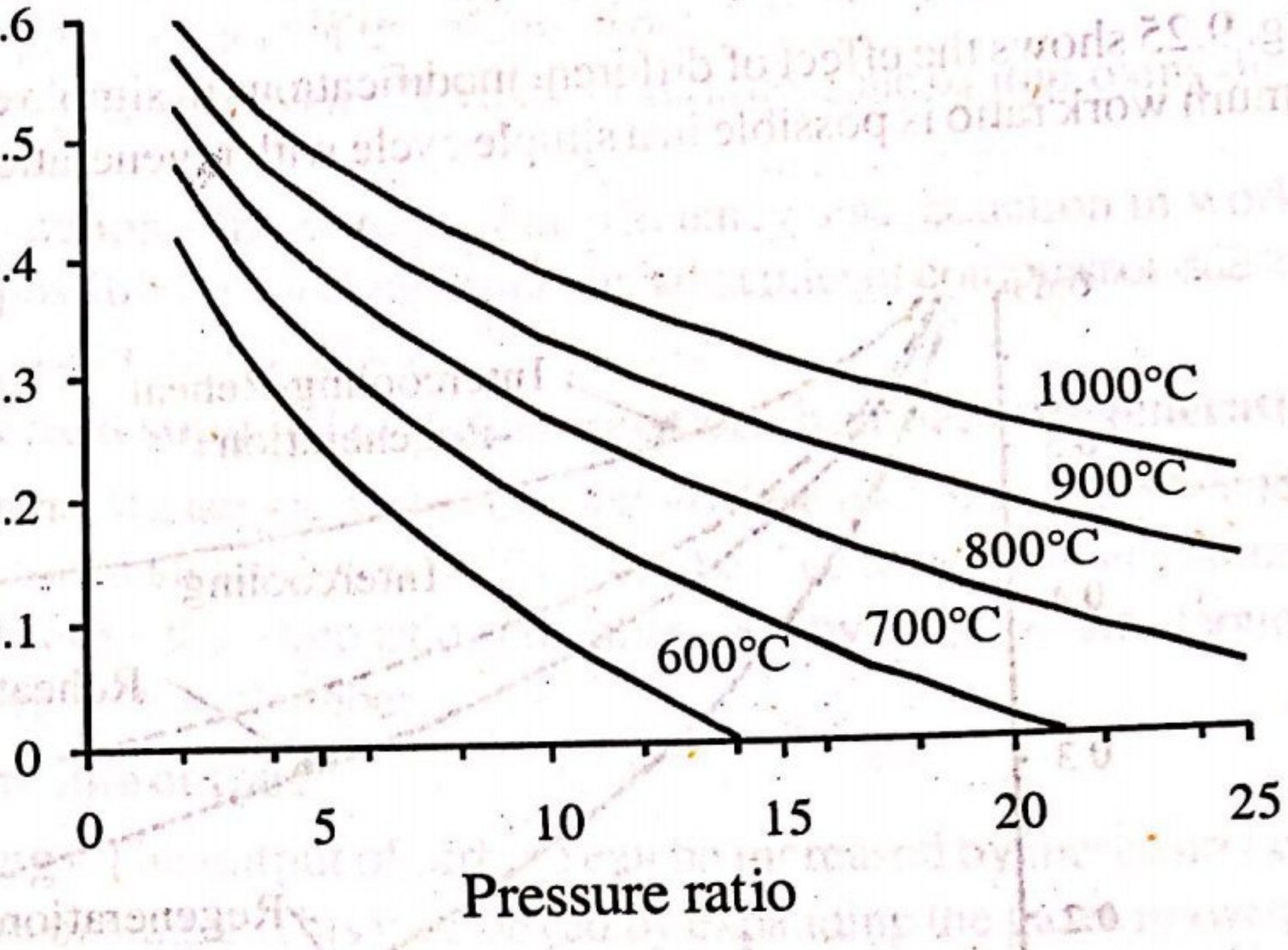


Fig. 9.23 Effect of turbine inlet temperature and pressure ratio on work ratio

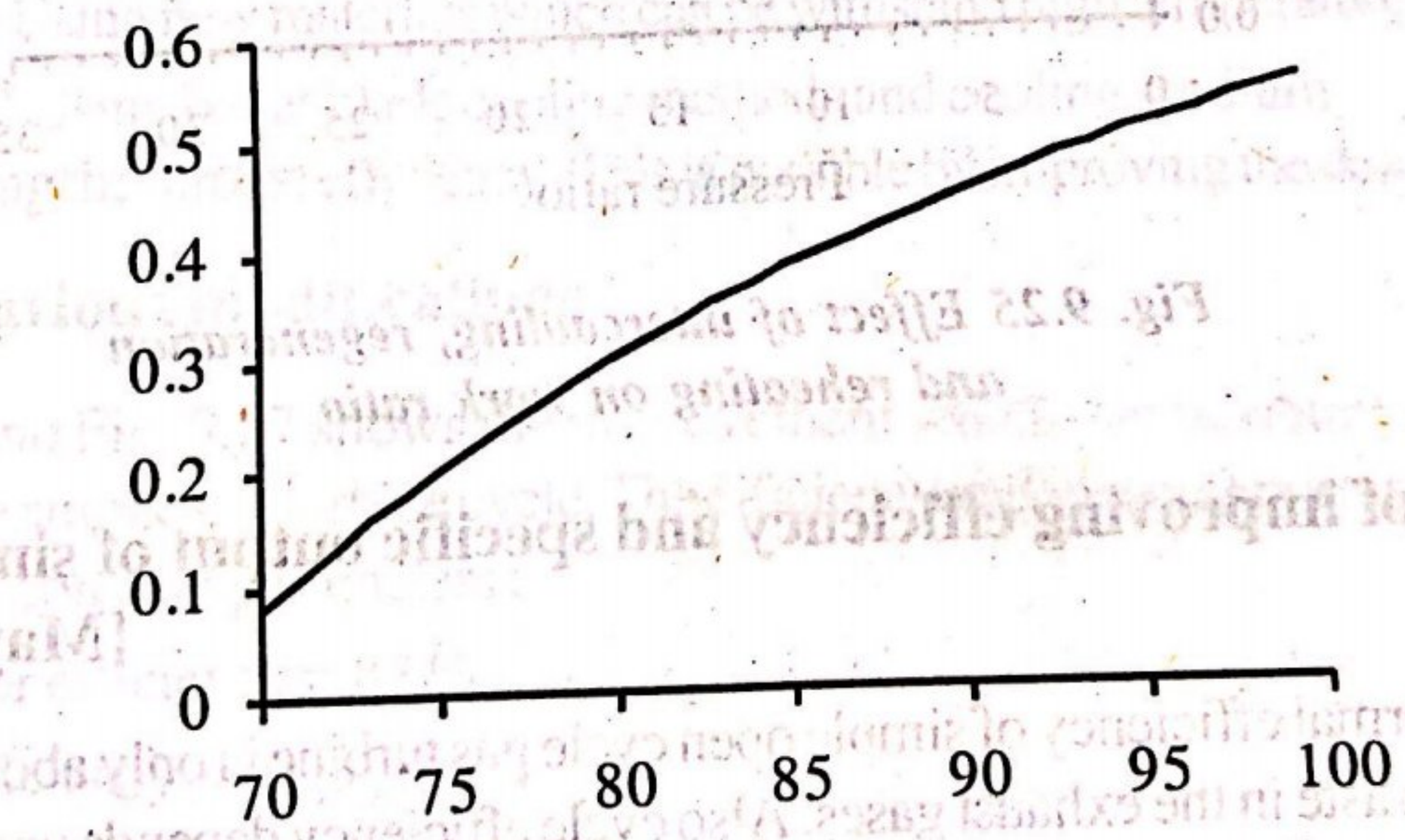


Fig. 9.24 Effect of compressor and turbine efficiencies on work ratio

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(4) Regeneration, intercooling and reheating

Fig. 9.25 shows the effect of different modifications in simple cycle on work ratio. The maximum work ratio is possible in a simple cycle with regeneration, intercooling and reheating.

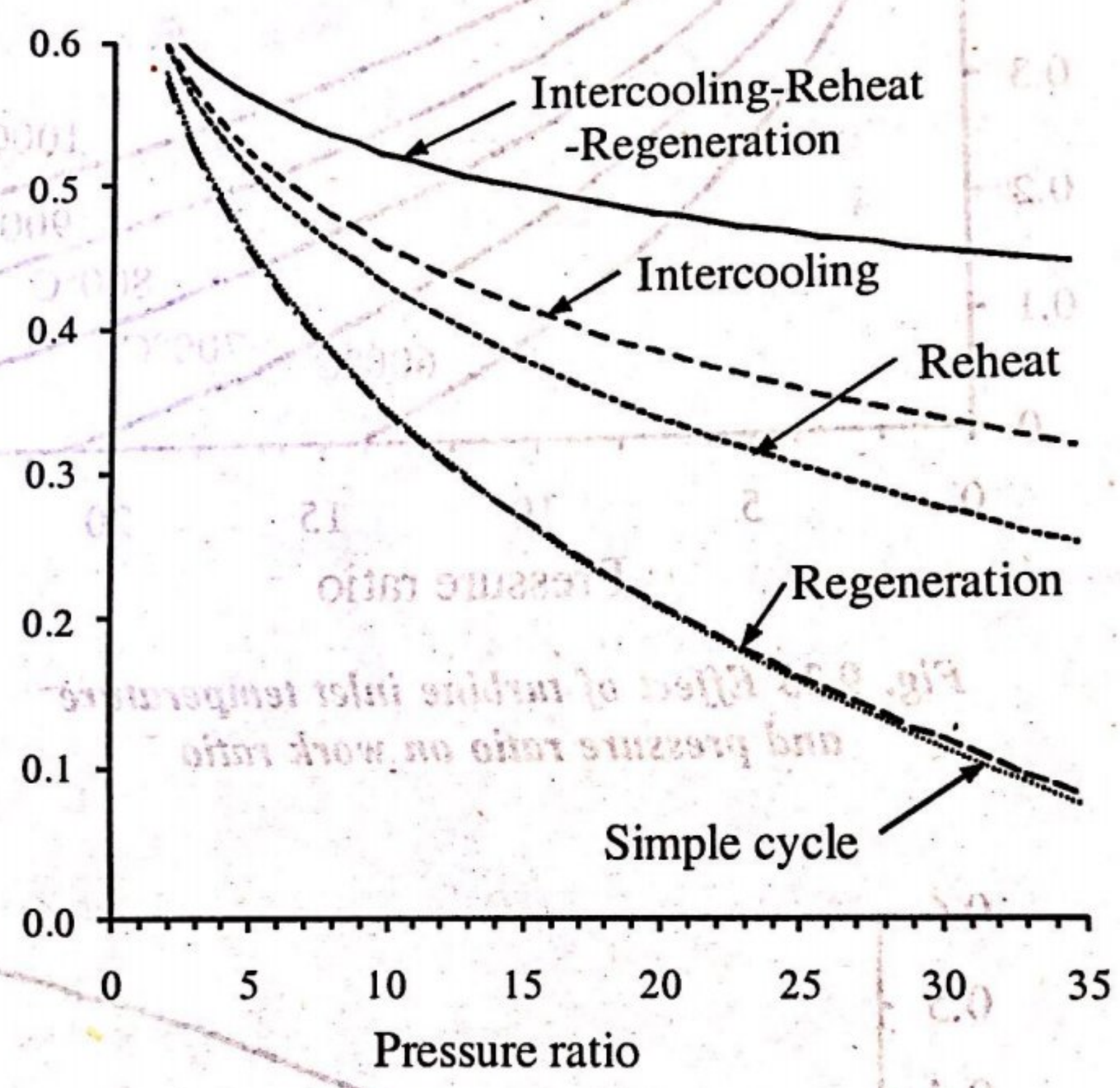


Fig. 9.25 Effect of intercooling, regeneration and reheating on work ratio

9.18 Means of improving efficiency and specific output of simple open cycle gas turbine [May '14, May '12]

The thermal efficiency of simple open cycle gas turbine is only about 16 to 23% as lot of heat energy waste in the exhaust gases. Also cycle efficiency depends upon the temperature of gases at turbine entry. Moreover, the metallurgical limitations do not permit the use of temperature of gases higher than about 1000 *K* for the simple gas turbine cycle. Hence, increase in efficiency cannot be possible through the increased temperature of the gases. However the thermal efficiency of a simple cycle gas turbine plant can be increased using the following methods :

(1) By reducing the work required to run the compressor

(a) **Intercooling** - Intercooling means the removal of heat from compressed air between the stages of compression. Therefore, work input in compression is reduced.

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(b) Lowering inlet air temperature of compressor - This method is not practical because this will increase the pressure ratio of the compressor.

(c) Increasing the compressor efficiency - This is possible by improving the design of compressor.

(d) Water injection - Improvement in efficiency and reduction in work input of compressor may be possible by injecting water into air at inlet of compressor due to the extra mass of water injected and cooling of air.

(2) By reducing the heat supplied in the combustion chamber - Regeneration

The turbine exhaust gases carry a large quantity of heat with them since their temperature is much above the ambient air temperature. They can be used to heat the air coming from the compressor before entering the combustion chamber, thereby reducing the amount of fuel to be burnt inside the combustion chamber.

(3) By improving turbine output

(a) **Reheating -** The output of turbine can be increased by increasing enthalpy of gases entering the lower stages. This is achieved by expanding the gases in two stages with reheater between the stages.

(b) Increasing the inlet turbine temperature

(i) Using better quality of fuel

(ii) Using new materials which can be withstand high temperature and pressure

(iii) Using better blade cooling methods and cooling medium

(c) Increasing the turbine efficiency -This is possible by improving the design of turbine.

9.19 Effect of various modifications

Fig. 9.26 and Fig. 9.27 shows the improvement which can be obtained by various modifications of the simple gas turbine cycle. The efficiency and work output curves are drawn based on following constant parameters :

Compressor efficiency = 85 %

Turbine efficiency = 86 %

Regenerator effectiveness = 0.75

Combustion efficiecy = 92%

Specific heat of air = 1.005 kJ/kg K and $\gamma = 1.4$

Specific heat of gases = 1.08 kJ/kg K and $\gamma = 1.33$

Calorific value of fuel = 40,000 kJ/kg

Atmospheric air temperature = 20°C

It should be noted that comparision in general in nature and not exact because of independence of many parameters which are dependent on the pressure ratio.

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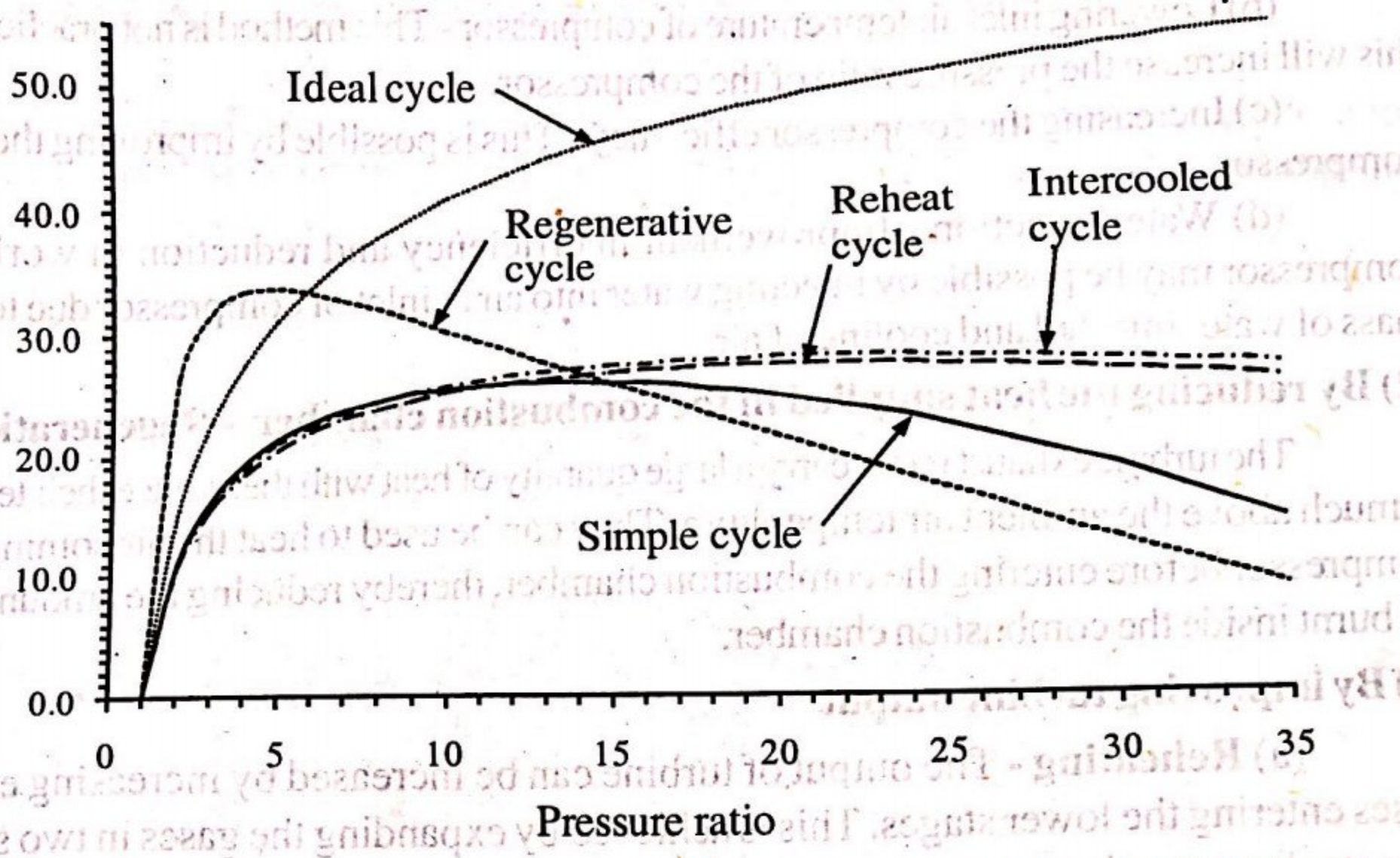


Fig. 9.26 Efficiency comparision of different improvements

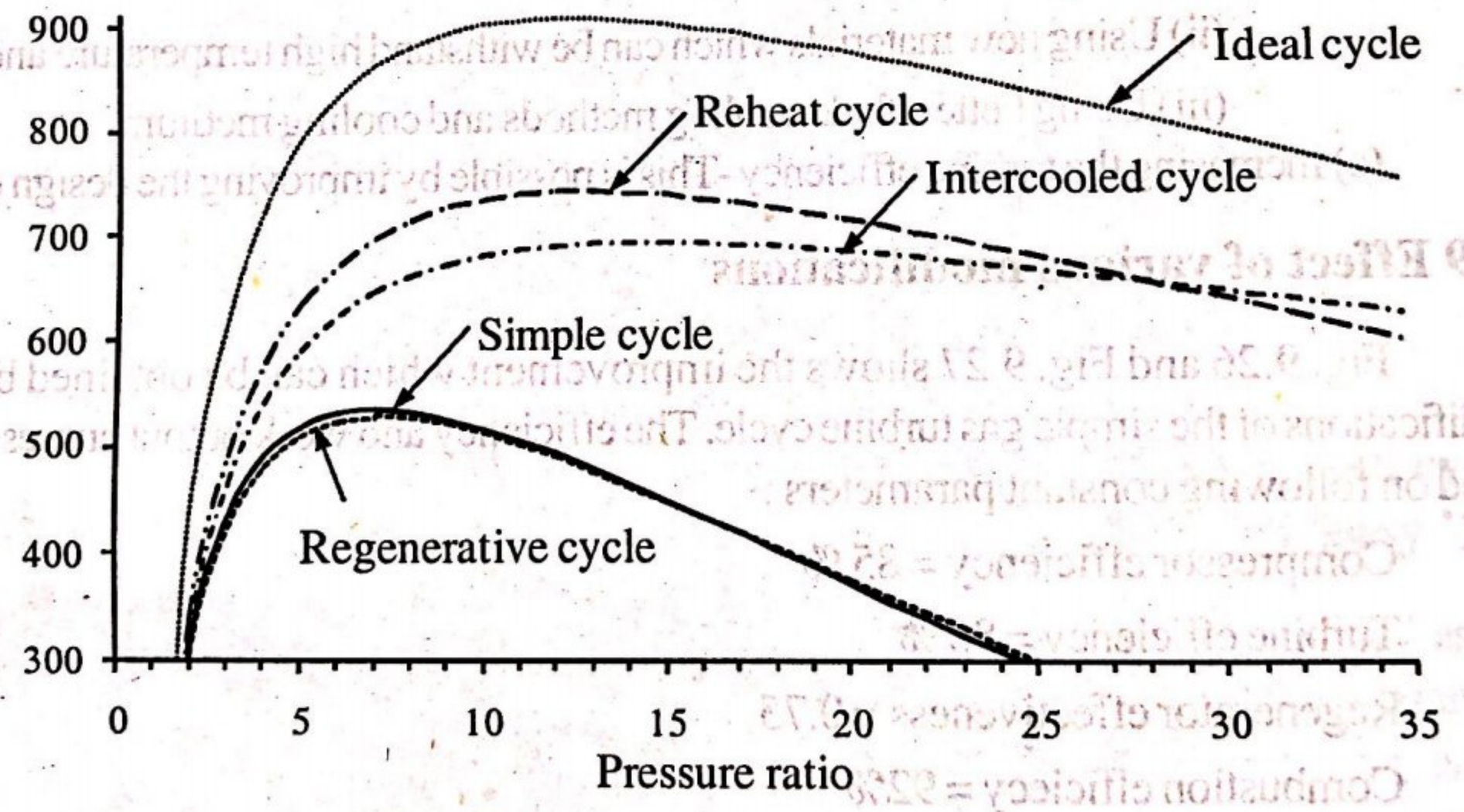


Fig. 9.27 Work output comparision of different improvements

Intercooling : The thermal efficiency of intercooled cycle is slightly less at low pressure ratio compared to simple cycle. However at high pressure ratio the efficiency of intercooled cycle is much greater than the simple cycle. The net work output of intercooled cycle is higher than that of simple cycle at all pressure ratios.

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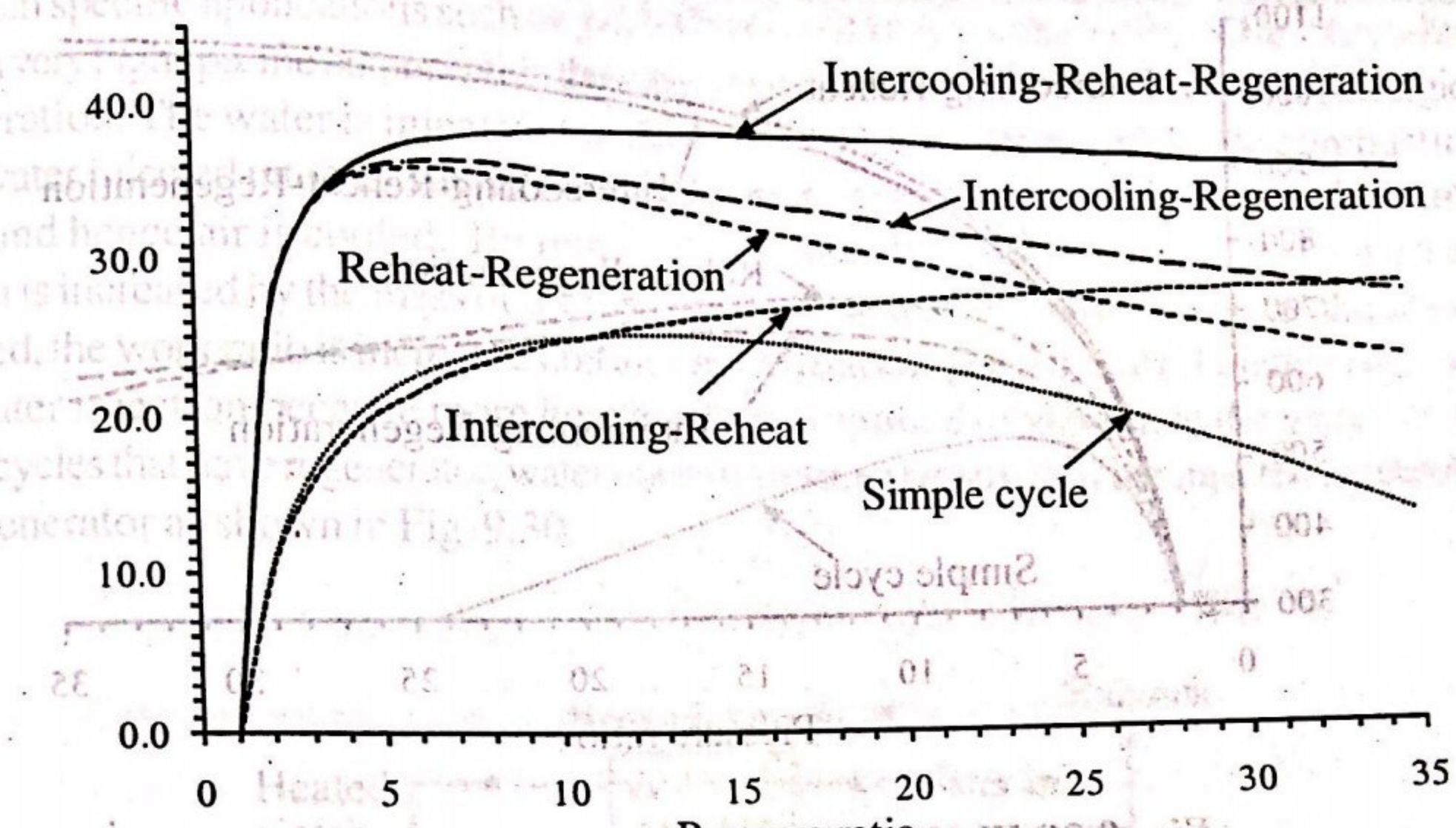


Fig. 9.28 Efficiency of different combinations

Regeneration : With use of regenerator in gas turbine plant, there is no variation in the net work output. But thermal efficiency at low pressure ratio is much greater than that of simple cycle. The efficiency of regenerative cycle reduces with increase in pressure ratio at high pressure ratio. If the pressure ratio is raised above certain limit, the temperature of compressed air is became more than exchaust gas temperature and hence regenerator cool the compressed air entering combustion chamber instead of heating. Hence regeneration is more benifited at low pressure ratios.

Reheating : At low pressure ratio, the efficiency of reheating cycle is lowered, but at high pressure ratios thermal efficiency is greater than that of simple cycle. The net work output produced by reheating cycle is higher than that of simple cycle at all pressure ratios.

The gas turbine cycle with different combinations (reheat - regeneration, intercooling-regeneration, reheat-intercooling, intercooling - reheat - regeneration) offers considerable increase in net work output and thermal efficiency as shown in Fig. 9.28 and Fig. 9.29. However, such a plant will become more complicated and costly and will need increased maintanance.

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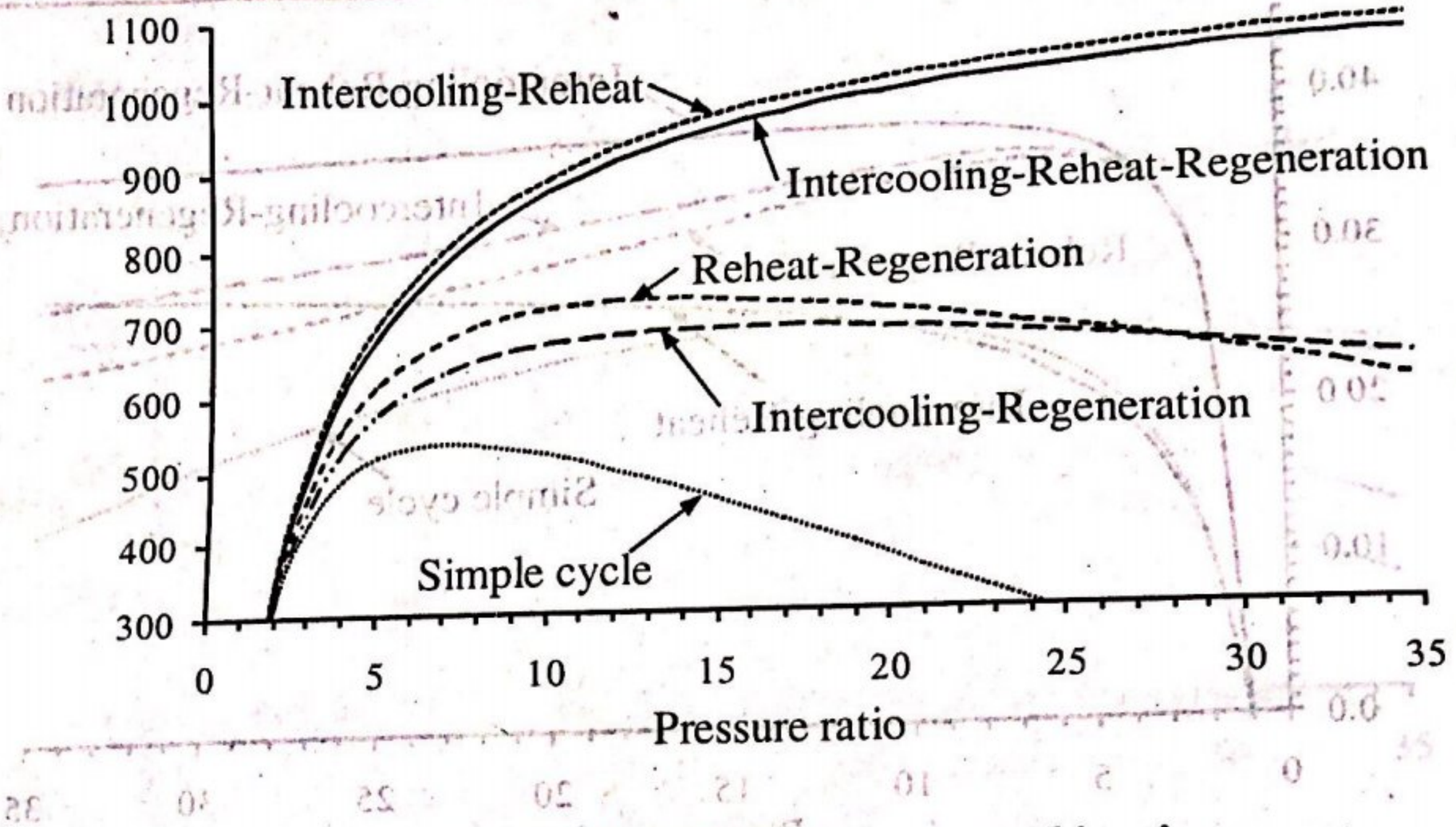


Fig. 9.29 Work output of different combinations

The effect of various modifications in simple cycle on thermal efficiency and work output at pressure ratio 8 are given in Table 9.1.

Table 9.1

Modification	Effect on work output	Effect on efficiency
1. Intercooling	25% (increase)	0.3% (decrease)
2. Regeneration	Nil	32% (increase)
3. Reheating	35% (increase)	3% (decrease)
4. Intercooling- regeneration	23% (increase)	51% (increase)
5. Intercooling-reheating	60% (increase)	3% (decrease)
6. Reheating-regeneration	32% (increase)	48% (increase)
7. Intercooling-reheating - regeneration	56% (increase)	59% (increase)

From above Table 9.1 and Fig. 9.26 to 9.29, it is cleat that the higher thermal efficiency is possible from compound cycle (simple cycle with intercooling, reheat, and regeneration). The maximum output is possible from simple cycle with intercooling and reheating. It is also clear that due to addition of intercooling, reheating and regeneration the peak value of thermal efficiency shifts towards lower pressure ratio, and larger range of pressure ratio for higher efficiency compared to simple cycle gas turbine plant.

9.20 Water injection

Water injection is one of the method of increasing power output of gas turbine. There

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are certain specific applications such as gas turbine used in naval torpedoes and aircraft which require a very high specific output within the maximum temperature limit imposed by metallurgical consideration. The water is injected into the combustion chamber before the combustion. When water injected into the compressed air the water evaporates, it absorbs latent heat from the air and hence air is cooled. By injection of water the total mass flow of the working medium is increased by the mass of the injected water and hence power output of the plant is increased, the work ratio is increased and air rate is reduced. The thermal efficiency decreases with water injection because more heat has to be supplied to vapourise the water. In gas turbine cycles that have regenerator, water injection is more beneficial if it is injected compressor and regenerator as shown in Fig. 9.30.

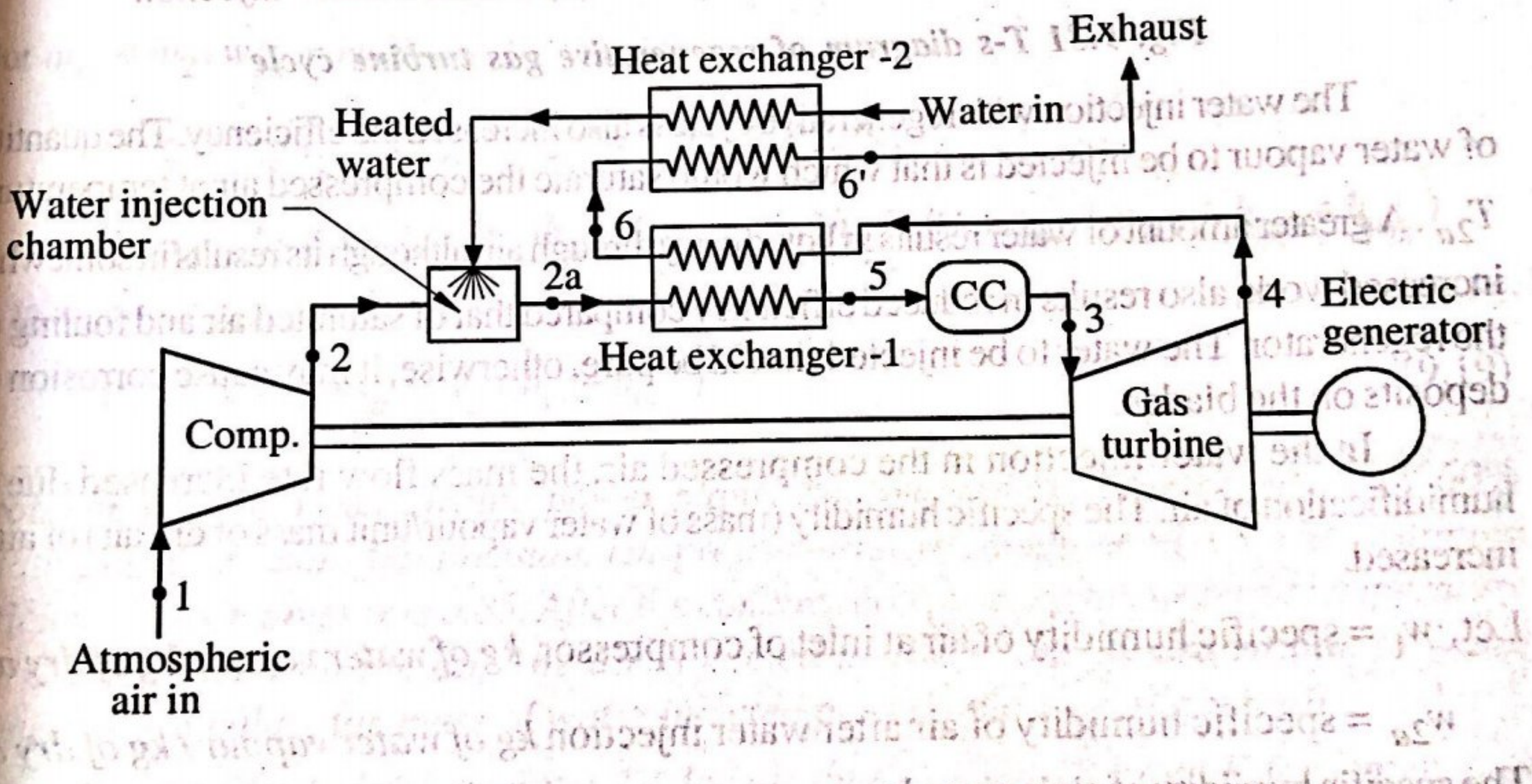
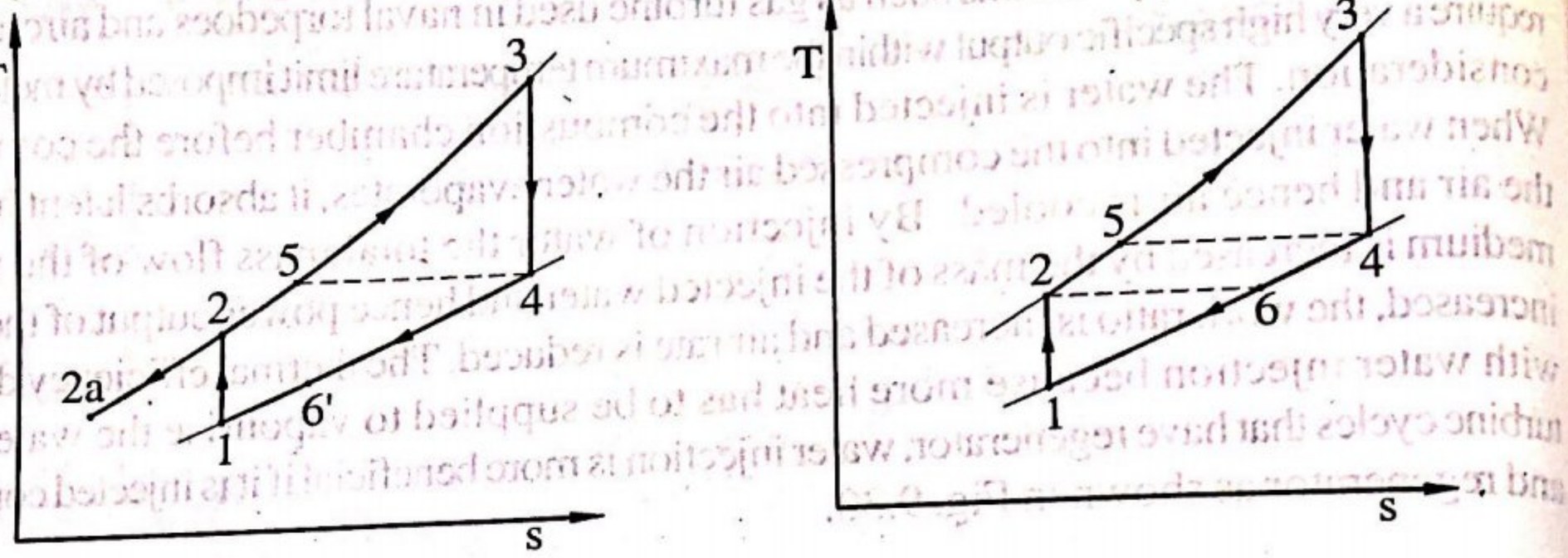


Fig. 9.30 Schematic diagram of GT plant with regenerator and water injection

The water to be injected first passes through the heat exchanger in which it is heated by exhaust gases. Fig. 9.31 shows the T-s diagrams of cycle with and without water injection. The water injected into the compressed air, and hence compressed air is cooled from state 2 to 2a at constant pressure. The mass flow rate of air is increased. In the regeneration air is heated from 2a to 5 instead of 2 to 5 (without water injection), and exhaust gases is cooled from 4 to $6'$ instead of 4 to 6 (without water injection).

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(a) with water injection *(a) without water injection*

Fig. 9.31 T-s diagram of regenerative gas turbine cycle

The water injection with regenerative cycle is also increased the efficiency. The quantity of water vapour to be injected is that which would saturate the compressed air at temperature T_{2a}. A greater amount of water results in liquid carry through air, although its results in somewhat increased work, also results in reduced efficiency compared that of saturated air and fouling of the regenerator. The water to be injected should be pure, otherwise, it may cause corrosion or deposits on the blades.

In the water injection in the compressed air, the mass flow rate increased due to humidification of air. The specific humidity (mass of water vapour/unit mass of dry air) of air is increased.

Let, w_1 = specific humidity of air at inlet of compressor, *kg of water vapour/kg of dry air*

w_{2a} = specific humidity of air after water injection *kg of water vapour / kg of dry air*

The specific humidity of air is given by

$$w = 0.622\left(\frac{p_v}{p - p_v}\right)$$

where p_v = partial pressure of water vapour

p = pressure of air vapour mixture

$$\therefore w_1 = 0.622\left(\frac{p_{v_1}}{p_1 - p_{v_1}}\right) \text{ and } w_{2a} = 0.622\left(\frac{p_{v_{2a}}}{p_{2a} - p_{v_{2a}}}\right)$$

where p_{v_1} = partial pressure of water vapour corresponding to state 1

$p_{v_{2a}}$ = partial pressure of water vapour corresponding to state $2a$.

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p_1 and p_{2a} = pressure of air-vapour mixture corresponding to state 1 and 2a respectively.

The increased in mass of water vapour in the air

$$w_{2a} - w_1 = 0.622\left(\frac{p_{v_{2a}}}{p_{2a} - p_{v_{2a}}} - \frac{p_{v_1}}{p_1 - p_{v_1}}\right) \quad ...(9.18)$$

Energy balance for water injection chamber

[Energy of dry air]$_2$ + [Energy of water vapour in air]$_2$ + [Energy of water injected]

= [Energy of dry air]$_{2a}$ + [Energy of water vapour in air]$_{2a}$

$$\therefore m_2 h_2 + w_2 m_2 h_{v_2} + m_w h_w = m_2 h_{2a} + w_{2a} m_2 h_{v_{2a}}$$

But $m_w = m_2 (w_{2a} - w_2)$

$$\therefore h_2 + w_2 h_{v_2} + (w_{2a} - w_2) h_w = h_{2a} + w_{2a} h_{v_{2a}}$$

$$\therefore h_{2a} - h_2 = (w_{2a} - w_2) h_w - (w_{2a} - w_2) h_{v_{2a}} \quad (\because \ p_{2a} = p_2 \ \therefore h_{v_2} = h_{v_{2a}})$$

$\therefore$ Enthalpy change of dry air

$$h_{2a} - h_2 = (w_{2a} - w_2)(h_w - h_{v_{2a}}) \quad ...(9.19)$$

Problem 9.10 *In a gas turbine power plant, the compressor takes air at 18°C, 60% relative humidity and 1 bar pressure. The pressure ratio of compression is 8. The isentropic efficiency of compressor is 0.85. After the compression process, the water at temperature 50°C is injected into the air. The air is cooled to 100°C and saturated due to water injection. Calculate the mass of water injected per unit mass of original air.*

Take, $C_p = 1.005$ *kJ/kg K and* $\gamma = 1.4$ *for air and* $C_p = 4.18$ *kJ/kg K for water.*

Solution : Given data : $T_1 = 18°C = 291$ K, $p_1 = 1$ bar, $\phi_1 = 60\%$ (relative humidity),

$p_2 = 8 \times 1 = 8$ bar, $\eta_c = 0.85$, $t_w = 50°C$, $T_{2a} = 100°C = 373$ K

From steam table, for $T_1 = 18°C$ saturated water vapour pressure $p_{s1} = 0.0206$ bar

The relative humidity of air is given by

$$\phi = \frac{p_{v1}}{p_{s1}} \qquad \therefore 0.6 = \frac{p_{v_1}}{0.0206} \qquad \therefore p_{v_1} = 0.01236 \text{ bar}$$

The specific humidity (kg of water vapour/kg of dry air) is given by

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$$w_1 = 0.622\left(\frac{p_{v_1}}{p_1 - p_{v_1}}\right)$$

$$= 0.622\left(\frac{0.01236}{1-0.01236}\right) = 0.00778 \text{ kg/kg of dry air}$$

The specific humidity of air at outlet of compressor is equal to specific humidity of air at inlet,

i.e. $w_2 = w_1$

For isentropic compression

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \qquad \therefore T_2 = 291(8)^{\frac{1.4-1}{1.4}} = 527.13\,K$$

Also, $\eta_c = \dfrac{T_2 - T_1}{T_2' - T_1}$

$$\therefore T_2' = T_1 + \frac{T_2 - T_1}{\eta_c} = 291 + \frac{527.13 - 291}{0.85} = 568.8\,K$$

Now, enthalpy of injected water

$$h_w = C_p T_w = 4.18 \times 50 = 209\,kJ/kg$$

Change in enthalpy of dry air due to water injection

$$h_{2a} - h_2 = C_p(T_{2a} - T_2')$$

$$= 1.005 \times (373 - 568.8) = -196.77\,kJ/kg$$

But $h_{2a} - h_2 = (w_{2a} - w_2)(h_w - h_{v_{2a}})$

From steam table, at 100°C, $h_{v_{2a}} = 2676$ kJ/kg and $P_{v_{2a}} = 1.0135$ bar

$$\therefore -196.77 = (w_{2a} - 0.00778)(209 - 2676)$$

$$\therefore w_{2a} = 0.0875 \text{ kg/kg of dry air}$$

Original mass of air $= m_1 + w_1 m_1$

$$= (1 + 0.00778)m_1 = 1.00778 m_1$$

Mass of air after water injection $= m_1 + w_{2a} m_1$

$$= (1 + 0.0875)m_1 = 1.0875 m_1$$

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Mass of water injected = $(1.0875 - 1.00778)\, m_1 = 0.07972\, m_1$

Mass of water injected/unit original mass of air

$$= \frac{0.07972 m_1}{1.00778 m_1} = 0.079 = \textbf{7.9\% of original air} \qquad \textit{Ans}$$

9.21 Combustion chambers [Nov. '14]

The combustion of a fuel is a chemical combination of the constituents (carbon and hydrogen) of fuel with oxygen as a result heat is produced. The main purpose of a gas turbine combustion is to introduce heat energy into the mass of air compressed by the compressor, by burning fuel in it so that the products of combustion can be expanded to get useful work output.

The design and development of a gas turbine combustion chamber is rather difficult compared to diesel engine due to continuously high combustion temperatures, large continuous flow and high heat energy release. One of the main problems associated with design of the gas turbine combustion chamber is to make steady and stable flame inside the combustion chamber.

The combustion chamber of gas turbine must fulfill the following **requirements** :

[May '14, May '13]

(1) **High combustion efficiency :** For good combustion efficiency the fuel must be atomized into small droplets and must be thoroughly mixed with the air. The inlet temperature and the turbulence required for mixing should be sufficient to minimize ignition delay and hence fuel-air mixture gets sufficient time for complete burning and uniform temperature distribution.

(2) **Minimum pressure losses :** The pressure losses occurs in the combustion chamber due to (i) turbulence and change in momentum of gases and (ii) skin friction between gases and combustion chamber walls. In order to reduce pressure losses, the diffuser is provided at entrace of combustion chamber and air is made stagnant or a reversal of flow if it is required.

(3) **Temperature at turbine inlet must be controlled :** The gas turbine requires about 100:1 air fuel ratio by weight. Therefore, it is impossible to ignite and maintain a continuous combustion with such weak mixture. It is necessary to provide rich mixture for ignition and continuous combustion, and therefore, it is required to allow necessary air (stoichiometric air fuel ratio- primary air) in the combustion zone, and remaining air (secondary air) must be added after complete combustion to reduce the gas temperature before passing into the turbine. The secondary air should be supplied in such a manner that a thorough mixing of air and products of combustion take place to give uniform temperature at the exist of the combustion chamber.

(4) **Stable continuous flame :** A stable continuous flame can be maintained inside the combustion chamber when the air stream velocity and fuel burning velocity are equal.

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Most of the fuels have low burning velocities, therefore special flame stabilization technique must be provided in the combustion chamber. The common methods of flame stabilization used in practice are bluff body method and swirl flow method.

(5) **A pilot or recirculated zone** should be created in the main flow to establish a stable flame which helps to ignite the combustion mixture continuously.

(6) **Complete combustion of the fuel :** Any unburnt fuel in the sense of the full heating value per kg not utilized, is directly reflected in the fuel consumption or thermal efficiency.

(7) Ignition must be reliable and accomplished with ease over wide range of atmosphere conditions, especially in aircraft applications.

(8) The volume and weight of the combustor must be kept within reasonable limits.

(9) Carbon deposits must not be formed under any expected condition of operation.

Types of combustion chamber :

The combustion chamber of gas turbine can be classified into three main types :

(1) Turbular or Can type

(i) Can type combustion with swirl flow flame stabilizer

(ii) Can type combustion with bluff-body flame stabilizer

(2) Annular type

(3) Turbo-annual or can-annular type

Can type of combustion chamber are widely used for industrial applications. However, it is not suitable for aircraft engines, due to large frontal area and weight. Can type combustion chamber is most suitable with centrifugal type compressor since diffuser divides the compressed air into channels. This gives good control of combustion.

Combustion is initiated by an electric spark and once fuel started burning, the flame is required to be stabilized. A recirculated or pilot zone is created in the main flow to establish a stable flame in order to sustain continuous combustion. Based on this flame stabilization techniques the Can type combustion chamber can be classified as swirl flow flame stabilizer and bluff-body flame stabilizer.

As shown in Fig. 9.32 a Can type combustor with swirl flow flame stabilizer, in which fuel is injected in downstream in the same direction as the air stream.

The can type combustor consists of diffuser, flame tube which is divided in three parts based on introduction of air as primary zone, secondary zone and tertiary (dilution) zone.

Diffuser : The function of diffuser is increases the static pressure and reduces the speed of air in order to diffuse it. Air from the compressor enters the combustion chamber at a high velocity, hence, there is necessity to reduce the velocity of air before entering the chamber so that the flame will remain alight through the range of operating condition. This is achieved with the help of a diffuser.

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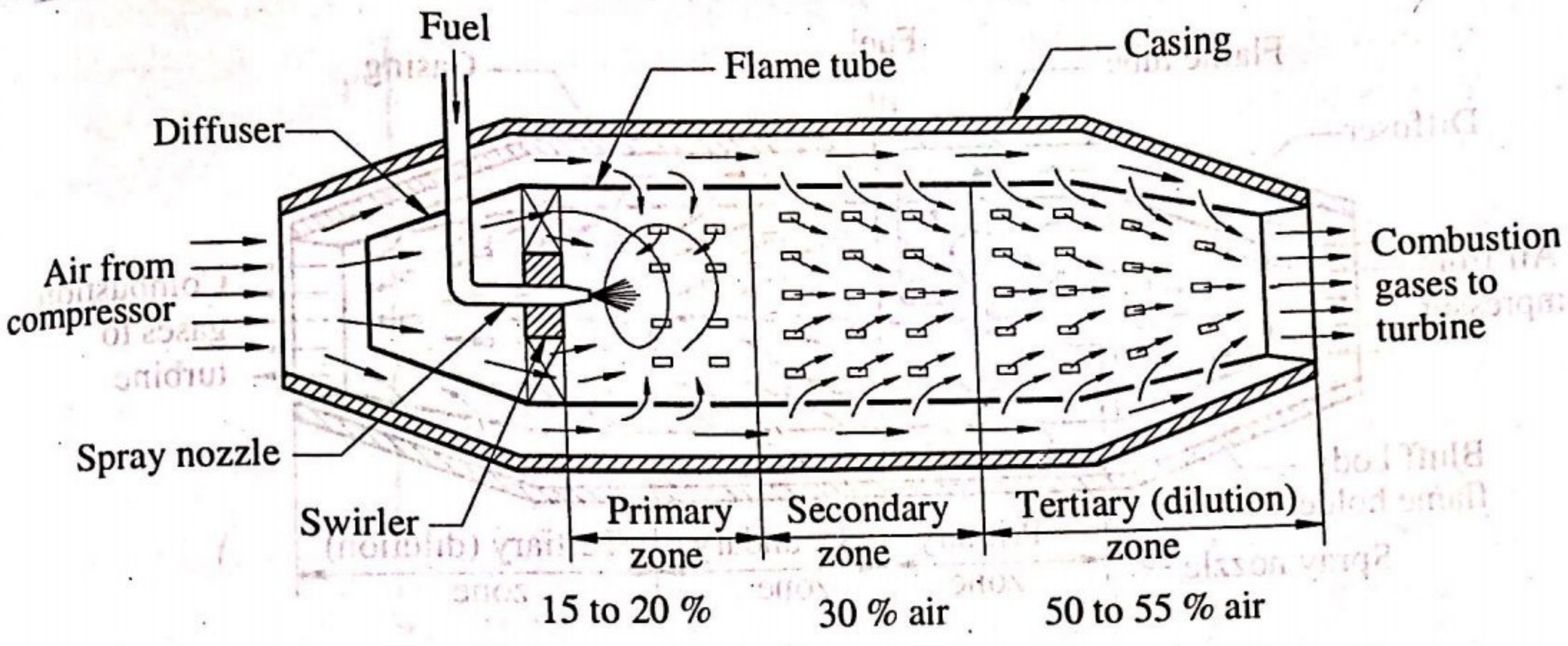


Fig. 9.32 Can type combustor with swirl flow flame stabilizer

Flame tube : As we know that the air fuel ratio in the combustion chamber (CC) is 100:1 while the stoichiometric ratio is about 15:1. Therefore, it is necessary to introduce the air in stages as per requirement. This is achieved by means of a flame tube which metering the air flow distribution along the chamber.

Primary zone: In the primary zone about 15 to 20% of total air from the compressor is directly fed through a swirler to the burner as primary air, to provide a rich fuel- air mixture (which is necessary for rapid combustion) in the primary zone which continuously burns, producing high temperature gases.

Swirler : Primary air flowing through the swirler (twisted radial vanes) produces a vortex motion creating a low pressure zone along the axis of the combustion chamber to cause reversal of flow. This is called recirculation of flow or flame stabilization.

Secondary zone : About 30% of total air is supplied through the holes in the flame tube in the secondary zone through the annulus round the flame tube to complete the combustion. The secondary air must be admitted at right points in the CC, otherwise the cold injected air may chill the flame locally thereby reducing the rate of reaction. The secondary air not only helps to complete the combustion process but also helps to cool the flame tube.

Tertiary zone : The remaining about 50% of total air is mixed with burnt gases in the tertiary zone to cool the gases down from 2000°C to the temperature required at inlet to the turbine about 1100°C which is suitable for turbine blade materials.

Fig. 9.33 shows a Can-type combustor with a bluff-body stabilizing flame. In this combustion chamber, the fuel is injected upstream into the air flow and a sheet metal cone and perforated baffle plate ensure the necessary mixing of fuel and air. The low pressure zone created downstream side causes the reversal of flow along the zones of the combustion chamber for uniform mixing and good combustion.

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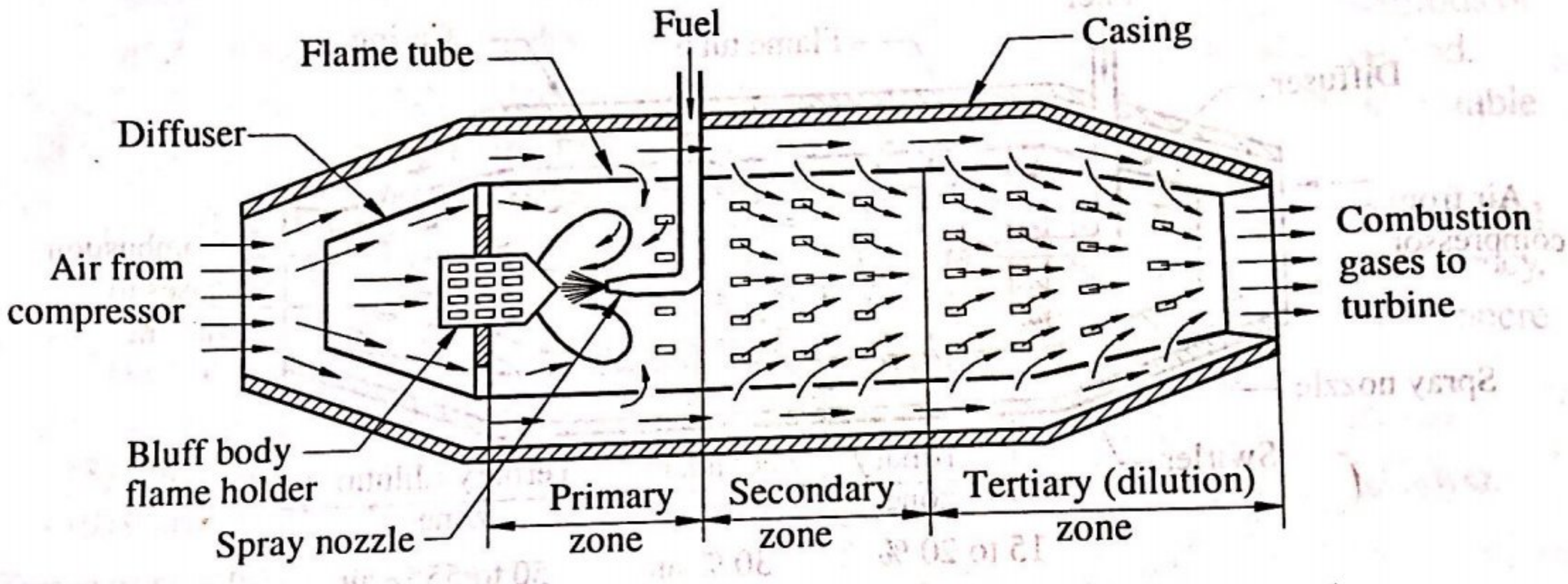


Fig. 9.33 Can type combustor with bluff-body flame stabilizer

9.22 Combined steam and gas turbine power plant [May '15, May '14, May '12]

As we know that gas turbine plants are generally used for supplying peak loads with other types of power plants (e.g. steam and hydro power plants) because gas turbine plant has a quick starting and good response characteristics. The temperature of gas turbine exhaust lies between 400 to 500°C and contains about 16% O_2 and 70% of initial quantity of energy carried away by the exhaust gases. Due to large exhaust loss the gas turbine plant has a low thermal efficiency (about 30%). Also, fuel used in gas turbine is costly, therefore cost of power generated by a gas turbine plant for utility system is high. To overcome its low cycle efficiency, a gas turbine plant may be used in conjuction with a steam turbine plant in an utility based load station to offer the advantages of gas turbine as quick starting and stopping, and permit flexible operation of combined plant over wide range of loads.

A combined cycle as the name implies is a combination of two cycles operating at different temperatures, each of which could operate independently. The heat rejected by the higher temperature cycle (Topping cycle) is recovered and used by lower temperature cycle (Bottoming cycle) to produced additional power to realize an improved overall efficiency. Topping cycle may Otto, Brayton or Rankine cycles. Bottoming cycle have all been Rankine cycle in all the cases.

In case of steam and gas turbine combined cycle, the topping plant operates on Brayton cycle and bottoming plant operates on Rankine cycle as shown in Fig. 9.34.

The exhaust gases coming from gas turbine has a temperature about 400-500°C, hence it carries large amount of energy. This energy goes as a waste if it is rejected to atmosphere. This waste heat energy may be utilized to produce steam in a Heat Recovery Steam Generator (HRSG) and may be expanded in the steam turbine to developed additional power. This combination is termed as steam and gas turbines ***combined cycle power plant***.

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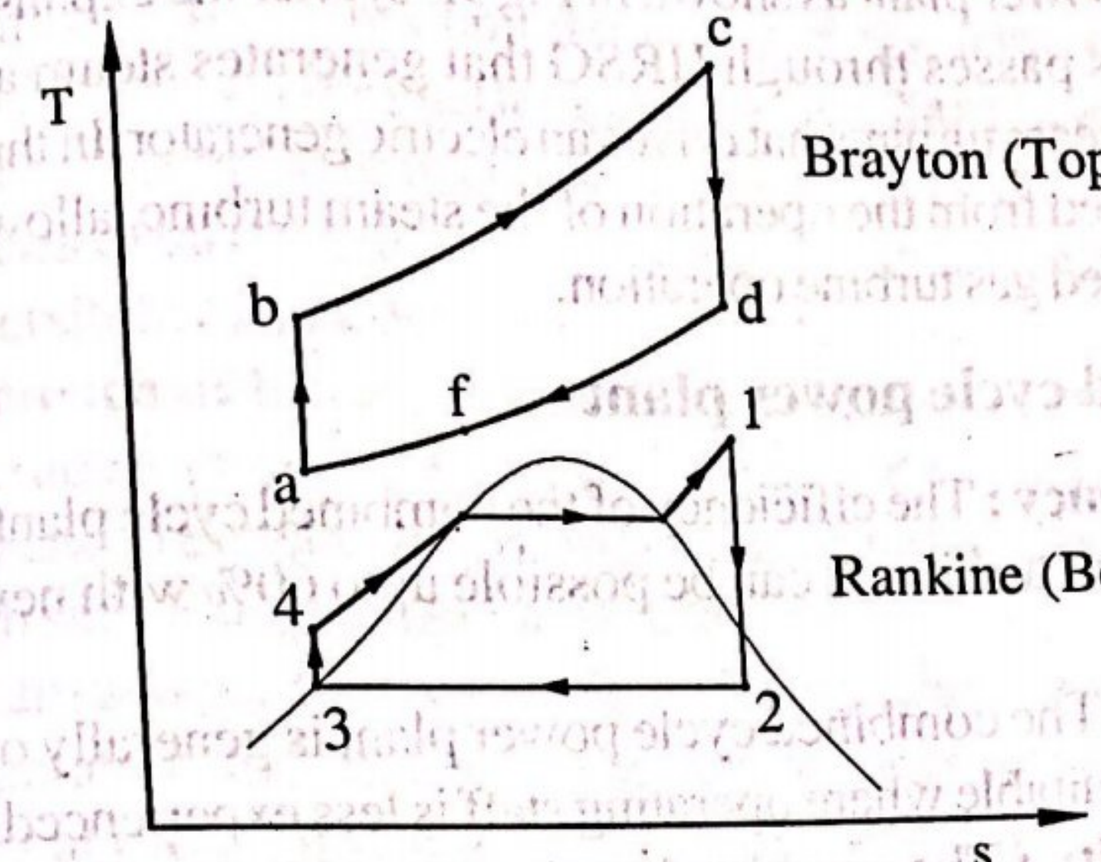


Fig. 9.34 T-s diagram of combined cycle (steam and gas turbine) power plant

This combined cycle efficiency ranges from 47 to 60% as compared to 30 to 40% of conventional steam or gas turbine power plant. It is desirable that the exhaust gases temperature of gas turbine should be above 570°C otherwise the steam cycle will be inefficient resulting in lower combined cycle efficiency.

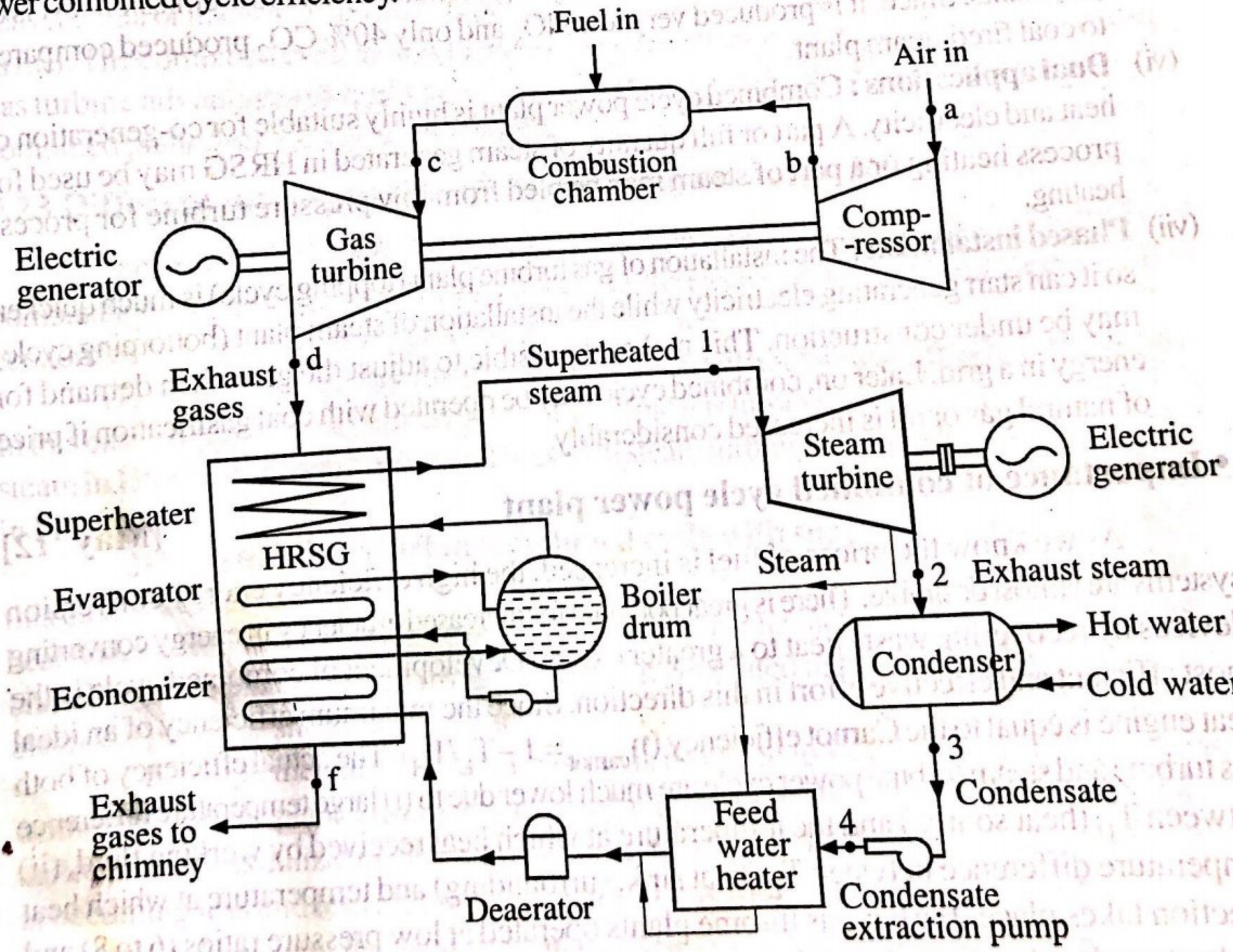


Fig. 9.35 Schematic diagram of combined cycle (Steam and gas turbine) power plant

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The combined cycle power plant as shown in Fig 9.35, after the expansion of gases in gas turbine the exhaust gases passes through HRSG that generates steam at one or more pressures. The steam is fed to steam turbine that drives an electric generator. In this arrangement the gas turbine can be decoupled from the operation of the steam turbine, allowing for steam turbine shutdown with continued gas turbine operation.

• Advantages of combined cycle power plant [May '15]

(i) **High overall plant efficiency :** The efficiency of the combined cycle plant is better than simple gas turbine or steam turbine. It can be possible upto 60% with new gas turbine technologies.

(ii) **Simplicity of operation :** The combined cycle power plant is generally operated fully automatically. Hence it is suitable where operating staff is less experienced.

(iii) **Great operating flexibility :** The combined cycle power plant makes it possible to start up and shutdown the plant quickly with less start-up losses.

(iv) **Less water required :** The amount of cooling water required is only about 40 to 50% that of steam power plant.

(v) **Less environmental impact :** Gas based combined cycle power plant may be suitable for use in heavy populated area due to their high efficiency and low emission levels of pollutants. Since, it is produced very less NO_x and only 40% CO_2 produced compared to coal fired steam plant.

(vi) **Dual applications :** Combined cycle power plant is highly suitable for co-generation of heat and electricity. A part or full quantity of steam generated in HRSG may be used for process heating or a part of steam may be bled from low pressure turbine for process heating.

(vii) **Phased installation :** The installation of gas turbine plant (topping cycle) is much quicker, so it can start generating electricity while the installation of steam plant (bottoming cycle) may be under construction. This makes it possible to adjust the growth in demand for energy in a grid. Later on, combined cycle may be operated with coal gasification if price of natural gas or oil is increased considerably.

• Importance of combined cycle power plant [May '12]

As we know the prices of fuel is increased, the high efficiency energy conversion systems are atmost desirable. There is great potential for increased efficiency in energy converting devices by recovering waste heat to a greater extent. Development of combined cycle is the most efficient and effective effort in this direction. Since the maximum efficiency of an ideal heat engine is equal to the Carnot efficiency ($\eta_{carnot} = 1 - T_L/T_H$). The actual efficiency of both gas turbine and steam turbine power cycle are much lower due to (i) large temperature difference between T_H (heat source) and the temperature at which heat received by working fluid, (ii) temperature difference between T_L (heat sink, surrounding) and temperature at which heat rejection takes place. Earlier gas turbine plants operated at low pressure ratios (6 to 8) and combustion chamber and turbine inlet temperatures were limited to 700°C due to the metallugical

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limit. Now with development of modern technology, it has become possible to use higher pressure ratio (20), and turbine inlet temperatures of 1200°C and even higher. The high pressure ratio and higher inlet turbine temperature increases the thermal efficiency. But, the temperature of exhaust gases of gas turbine is much greater than atmosphere. Therefore, there is great thermal irreversibility and a decrease of availability because the heat transfer (rejection) to atmosphere through such a large temperature difference. Also, in case of steam turbine cycle, steam temperature in cycle does not exceed 600°C, and hence there is great thermal irreversibility and a decrease of availability because of heat transfer from combustion gases to steam through such a large temperature difference.

If steam and gas turbines cycle works together, then it is possible to reduce the heat transfer temperature differences, as gas turbine cycle works at high temperature and steam turbine works at low temperature. Therefore, the heat supplied to gas turbine is possible at higher temperature and heat rejection takes place in condenser at surrounding temperature. By superimposing a high temperature gas turbine plant (topping cycle) to steam turbine plant (bottoming cycle), a higher energy conversion efficiency from fuel to electricity and minimum heat loss can be achieved. Hence thermal efficiency of combined cycle is increased to a great extent. Combined cycle power plants are gaining increasing acceptance as alternatives to conventional or nuclear cycle due to high thermal efficiency as high as 60% utilising natural gas as fuel. The combined cycle power plant not only increases efficiency, also offers utilities the gas turbine advantages of quick starting and stopping and permit flexible operation of the combined plant over a wide range of loads.

9.23 Different arrangements of combined cycle power plant

The gas turbine, HRSG and steam turbine can be arranged in many different combinations depending on size of the gas turbine, the electric power generation requirements, economics of applications and co-generation as follows :

(1) Steam and gas turbine combined cycle without supplementary fuel firing : In this case, the heat in the exhaust gases of a simple gas turbine plant can be used to generate steam in HRSG. The steam generated is used in steam turbine for power generation as shown in Fig. 9.35.

(2) Steam and gas turbine combined cycle with supplementary fuel firing : In this case, the heat and O_2 carried with the exhaust gases both are passes through the waste heat boiler in which the heat added to gases by supplying the fuel in the secondary combustion chamber (supplementary fuel firing). The steam generated in the boiler is used to produced electrical power as shown in Fig. 9.36.

The supplementary fuel firing system is provided in HRSG to produce more power from steam turbine plant. However, supplementary fired combined cycle is less efficient than unfired due to less utilization of waste and supplementary heat. Therefore supplementary firing is becoming less and less attractive. Generally it is more profitable to burn the fuel in the combustion chamber of the gas turbine plant itself since the heat is supplied to the system at a

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temperature higher than that in the steam cycle.

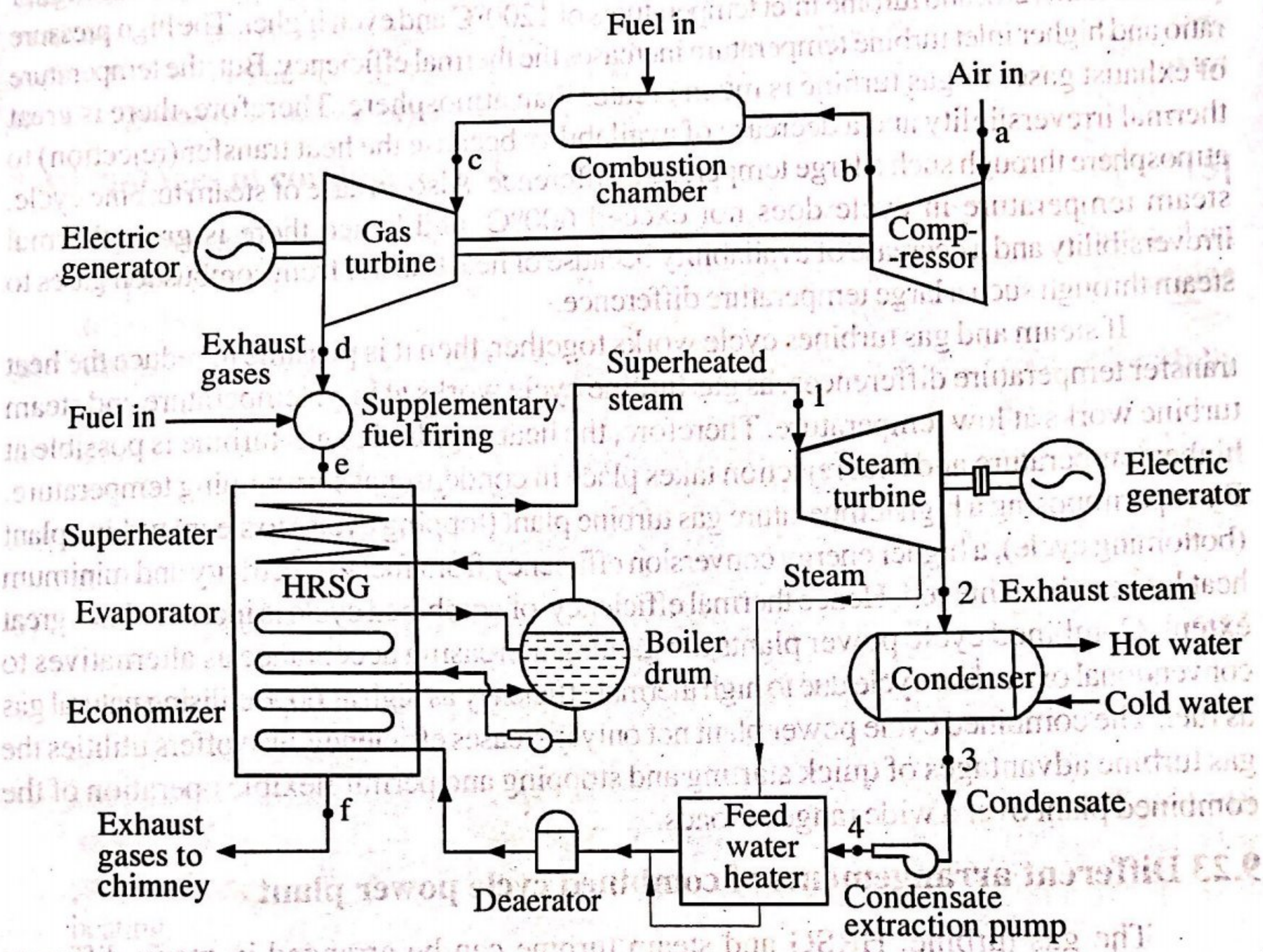


Fig. 9.36 Steam and gas turbine cycle with supplementary fuel fired plant

(3) Steam and gas turbine combined cycle with process heating : In this case, the heat in the exhaust gases of gas turbine plant is used to generate the low pressure steam in HRSG. In addition to HRSG, the steam boiler is used to produce high pressure and temperature steam. This high pressure steam and low pressure steam from HRSG are expanded in non condensing type steam turbine to produced the electricity. The exhaust steam coming from steam turbine is used as process steam as shown in Fig. 9.37. This type of co-generation is most economically used in process industries such as chemical, steel, paper and petroleum refining. This arrangement has a considerable advantage as it does not require steam condensing system with its capital costs. It also minimizes, the impact of heated discharges to the environment.

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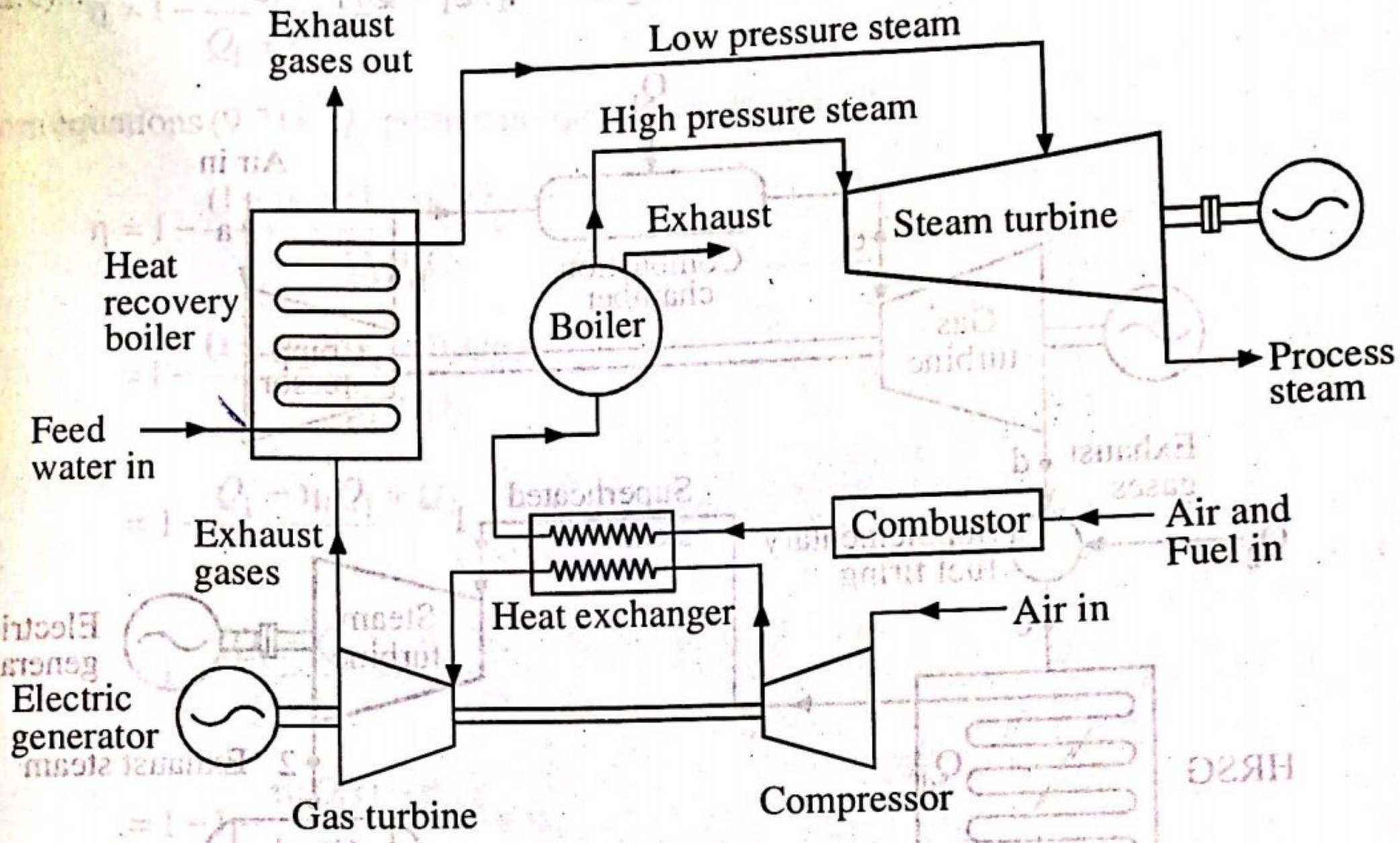


Fig. 9.37 Combined cycle power plant with process heating

9.24 Thermodynamic analysis of combined cycle

Let us consider two cyclic power plants coupled in series with supplementary fuel firing, the topping plant operating on Brayton cycle and the bottoming one operating on Rankine cycle as shown in Fig. 9.38.

Let, Q_1 = heat supplied to the combustion chamber of gas turbine plant

Q_d = heat rejected by gas turbine to HRSG

Q_2 = heat supplied to HRSG through the supplementary fuel firing

Q_c = heat rejected by condenser to cooling water

η_1 = efficiency of topping cycle

η_2 = efficiency of bottoming cycle

η = overall efficiency of combined cycle

Efficiency of topping (Brayton) cycle

$$\eta_1 = 1 - \frac{Q_d}{Q_1} \qquad \therefore Q_d = (1 - \eta_1) Q_1 \qquad \ldots(9.20)$$

Efficiency of bottoming (Rankine) cycle

$$\eta_2 = 1 - \frac{Q_c}{Q_d + Q_2} \qquad \therefore Q_c = (1 - \eta_2)(Q_d + Q_2)$$

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$$\therefore Q_c = (1-\eta_2)[(1-\eta_1)Q_1 + Q_2] \quad ...(9.21)$$

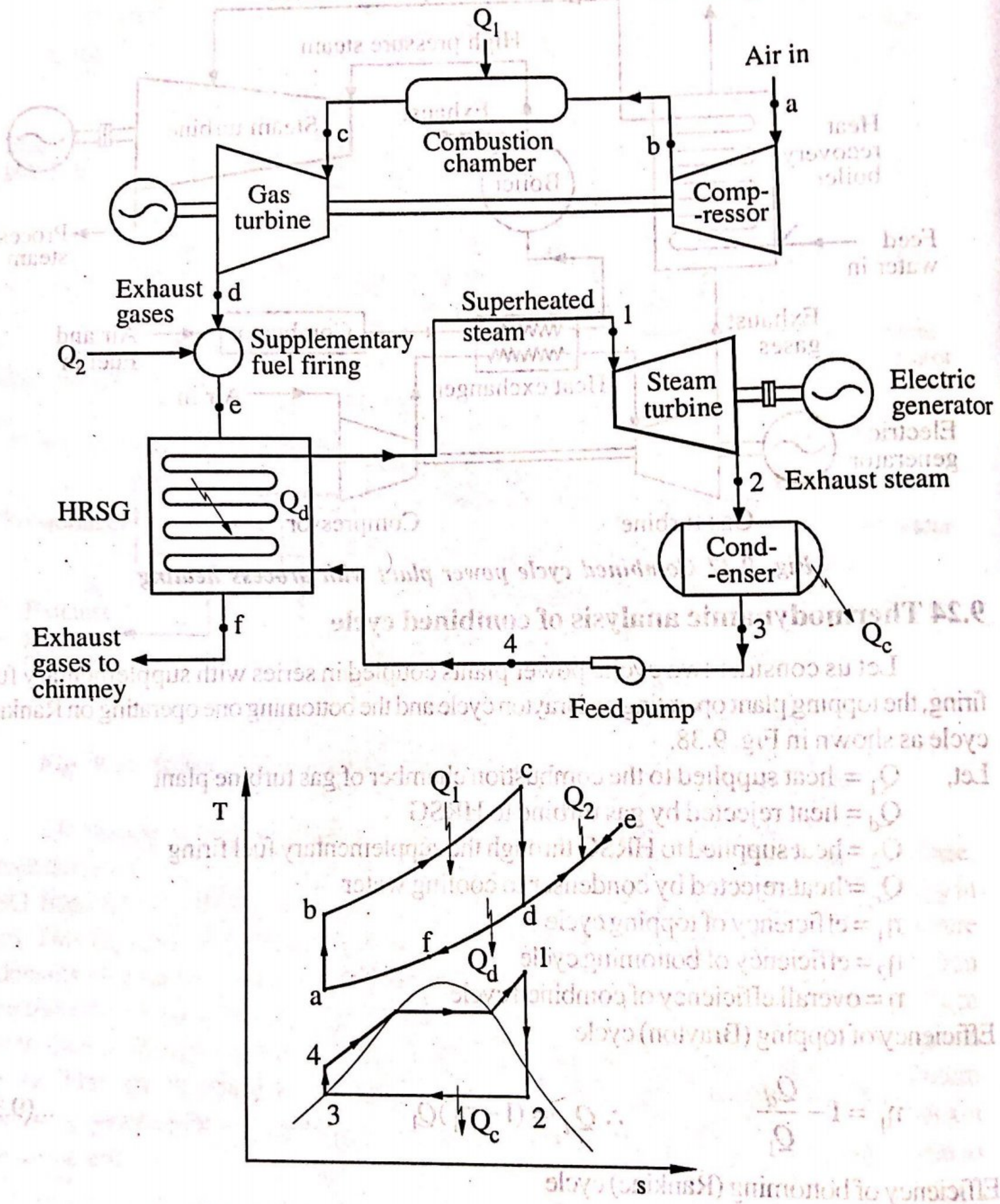


Fig. 9.38 Combined cycle power plant with supplementary firing

Efficiency of combined cycle

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$$\eta = 1 - \frac{Q_c}{Q_1 + Q_2} \qquad ...(9.22)$$

From equations (9.21), Q_c putting in equation (9.22) we get,

$$\eta = 1 - \frac{(1-\eta_2)[(1-\eta_1)Q_1 + Q_2]}{Q_1 + Q_2}$$

$$= 1 - \frac{(1-\eta_2)[Q_1 - \eta_1 Q_1 + Q_2]}{Q_1 + Q_2}$$

$$= 1 - \frac{Q_1 - \eta_1 Q_1 + Q_2 - \eta_2 Q_1 + \eta_1 \eta_2 Q_1 - \eta_2 Q_2}{Q_1 + Q_2}$$

$$= 1 - \frac{(Q_1 + Q_2) - \eta_1 Q_1 (1-\eta_2) - \eta_2 (Q_1 + Q_2)}{Q_1 + Q_2}$$

$$= 1 - 1 + \frac{\eta_1 Q_1 (1-\eta_2)}{Q_1 + Q_2} + \eta_2$$

$$\boxed{\eta = \eta_2 + \frac{\eta_1 Q_1 (1-\eta_2)}{Q_1 + Q_2}} \qquad ...(9.23)$$

If $Q_2 = 0$, No supplementary fuel firing, then

$$\eta = \eta_2 + \eta_1 (1 - \eta_2)$$

This is efficiency of a combined cycle without secondary fuel firing.

In the combined cycle power plant, even if individual efficiencies are low, it is possible to achieve a fairly high combined cycle efficiency, which cannot be attained by a single cycle.

For example, if $\eta_1 = 0.3$ and $\eta_2 = 0.35$

$$\therefore \ \eta = 0.35 + 0.3(1 - 0.35) = 0.545 = \mathbf{54.5\%}$$

Consider m_a kg/s of air entering to the compressor.

Ideal compressor work $W_C = m_a C_{pa} (T_b - T_a)$

Actual compressor work $W_C = m_a C_{pa} (T_b' - T_a) = \dfrac{m_a C_{pa} (T_b - T_a)}{\eta_c}$

where η_c = compressor efficiency $= \dfrac{T_b - T_a}{T_b' - T_a}$

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Ideal heat supplied to combustion chamber

$$Q_1 = m_{f1} \times C.V. = (m_{f1} + m_a) C_{pg} (T_c - T_b')$$

Actual heat supplied to combustion chamber

$$Q_1 = m_{f1} C.V. \times \eta_{comb} = (m_{f1} + m_a) C_{pg} (T_c - T_b')$$

Ideal work produced by gas turbine

$$W_T = (m_a + m_{f1}) C_{pg} (T_c - T_d)$$

Actual work produced by gas turbine

$$W_{GT} = (m_a + m_{f_1}) C_{pg} (T_c - T_d) \times \eta_t$$

where, η_t = turbine efficiency $= \dfrac{T_c - T_d'}{T_c - T_d}$

Heat supplied to the HRSG for supplementary fuel firing

$$Q_2 = m_{f_2} \times CV \times \eta_{comb}$$

$$= (m_a + m_{f_1} + m_{f_2}) C_{pg} (T_e - T_d)$$

The temperature and pressure of steam coming from HRSG depends upon the temperature of gas turbine exhaust entering into HRSG. The steam cycle efficiency is satisfactory only when the gas turbine exhaust temperature is above 570°C.

Considering the heat balance in the HRSG

$$(m_a + m_{f_1} + m_{f_2}) C_{pg} (T_e - T_a) = m_s (h_1 - h_4)$$

where m_s = mass flow rate of steam

Work produced by steam turbine

$$W_{ST} = m_s (h_1 - h_2)$$

Work supplied to the pump

$$W_p = m_s (h_4 - h_3)$$

Net work produced by combined cycle power plant

$$W_{net} = W_{net\,GT} + W_{net(ST)}$$

$$= W_{GT} - W_C + W_{ST} - W_P$$

Thermal efficiency of combined cycle

$$\eta = \frac{W_{net}}{Q_1 + Q_2}$$

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Problem 9.11 *In a combined cycle power plant, compressor takes 1500 tones air per hour at 26°C and 1 bar pressure. The maximum temperature of gas turbine is limited to 850°C. The pressure ratio is 8. The exhaust gases coming from gas turbine is heated further to 700°C before entering into heat recovery steam generator. The steam is generated at 50 bar and 500°C. The exhaust gas temperature is limited to 200°C to avoid the condensation of corrosive gases. The condenser pressure is 0.07 bar. Assume isentropic compression in compressor and isentropic expansion in both turbines. Calculate (i) Power generated in each unit of the cycle, (ii) Specific fuel consumption, (iii) Thermal efficiency of each cycle and combined cycle.*
Take, $C_p = 1$ kJ/kg K, $\gamma = 1.4$ for both air and gas, C.V of fuel used = 41,000 kJ/kg

Solution : Given data : $m_a = 1500$ tones/hr $= 416.67$ kg/s, $T_a = 26°C = 299$ K, $p_a = 1$ bar, $T_c = 850°C = 1123$ K, $p_b/p_a = 8$, $T_f = 200°C = 473\,K$, $\eta_c = \eta_{GT} = \eta_{ST} = 100\%$, $C_{pa} = C_{pg} = 1$ kJ/kg K, $\gamma = 1.4$, CV = 41,000 kJ/kg, $P_{GT} = ?$, $P_{ST} = ?$, $sfc = ?$, $\eta_{th(GT)} = ?$, $\eta_{th(ST)} = ?$, $\eta_o = ?$

Gas turbine cycle :

For process a-b, $\dfrac{T_b}{T_a} = \left(\dfrac{p_b}{p_a}\right)^{\frac{\gamma-1}{\gamma}}$ $\quad \therefore T_b = 299 \times (8)^{\frac{1.4-1}{1.4}} = 541.62$ K

For process c-d, $\dfrac{T_c}{T_d} = \left(\dfrac{p_b}{p_a}\right)^{\frac{\gamma-1}{\gamma}}$ $\quad \therefore T_d = 1123 \times \left(\dfrac{1}{8}\right)^{\frac{1.4-1}{1.4}} = 619.94$ K

For process b-c, assuming ideal condition,

Heat generated in CC-I = Heat gain by gases

$$\therefore m_{f_1} \cdot CV = (m_a + m_{f_1}) C_{pg} (T_c - T_b)$$

$$\therefore m_{f_1} \cdot 41,000 = (416.67 + m_{f_1}) \times 1 \times (1123 - 541.62)$$

$$\therefore m_{f_1} = 5.99 \text{ kg/s} = 21.56 \text{ tones/hr}$$

For process d-e, assuming ideal condition,

Heat generated in CC-II = Heat gain by exhaust gases

$$\therefore m_{f_2} \cdot CV = (m_a + m_{f_1} + m_{f_2}) \cdot C_{pg} (T_e - T_d)$$

$$\therefore m_{f_2} \times 41,000 = (416.67 + 5.99 + m_{f_2}) \times 1 \times (973 - 619.94)$$

$$\therefore m_{f_2} = 3.67\,kg/s = 13.21 \text{ tones/hr}$$

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Total mass of fuel supplied to the plant $m_f = m_{f_1} + m_{f_2} = 21.56 + 13.21 = 34.77$ tones/hr

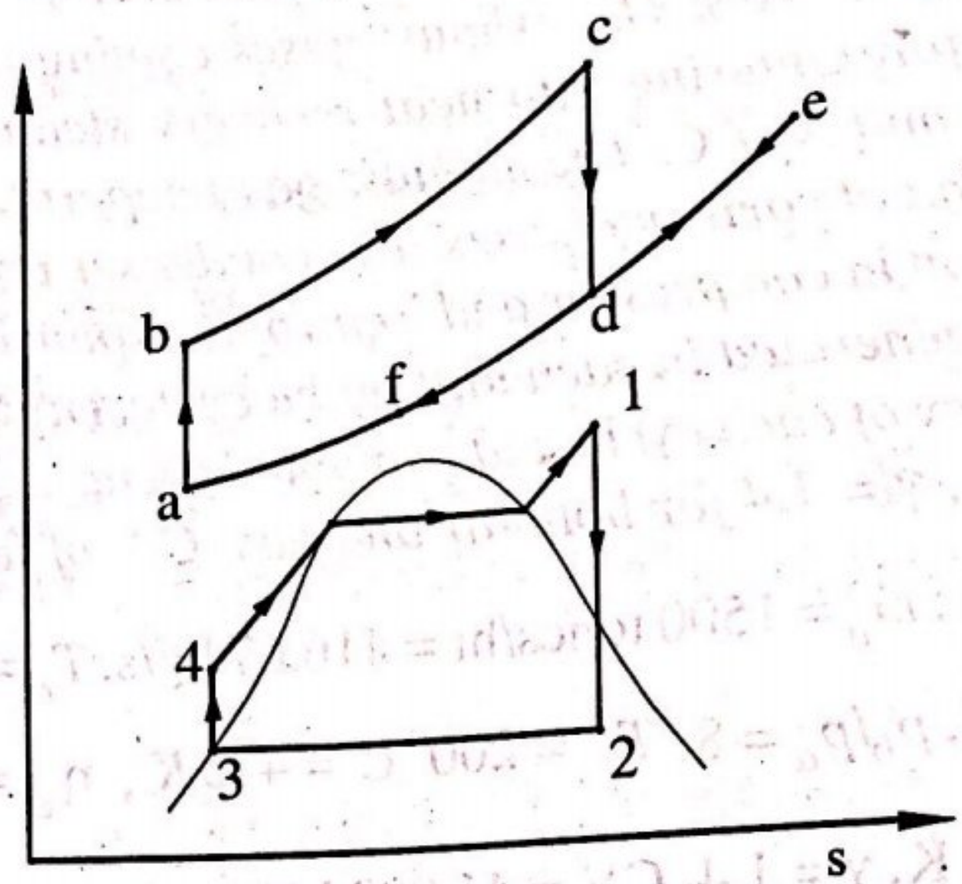


Fig. 9.39

Power developed by gas turbine plant

$$P_{GT} = W_{GT} - W_C$$

$$= (m_a + m_{f_1}) \times C_{pg} \times (T_c - T_d) - m_a \times C_{pa} \times (T_b - T_a)$$

$$= (416.67 + 5.99) \times 1 \times (1123 - 619.94) - 416.67 \times 1 \times (541.62 - 299)$$

$$= 212623.34 - 101092.47 = \mathbf{111530.86\ kW} \quad \textit{Ans}$$

Thermal efficiency of gas turbine plant

$$\eta_{th(GT)} = \frac{P_{GT}}{m_{f_1} \times CV} = \frac{111530.86}{5.99 \times 41,000} \times 100 = \mathbf{45.41\ \%} \quad \textit{Ans}$$

Steam turbine cycle :

From superheated steam table, at 50 bar and 500°C

$$h_1 = 3433.7 \text{ kJ/kg}, \; s_1 = 6.977 \text{ kJ/kg K}$$

From saturated steam table, at 0.07 bar

$$h_{f_2} = 163.4\, kJ/kg, \; h_{fg_2} = 2409.1\, kJ/kg$$

$$s_{f_2} = 0.559\, kJ/kg, \; s_{fg_2} = 7.717\, kJ/kg\, K$$

For process 1-2, $s_1 = s_2$ $\quad \therefore s_1 = s_{f_2} + x_2 s_{fg_2}$

$$\therefore 6.977 = 0.559 + x_2(7.717) \quad \therefore x_2 = 0.83$$

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Also, $h_2 = h_{f_2} + x_2 h_{fg_2} = 169.4 + 0.83 \times (2409.1) = 2172.9$ kJ/kg

Neglecting pump work, $h_4 = h_3 = h_{f2} = 163.4\, kJ/kg$

Heat balance in HRSG, assuming ideal condition

Heat rejected by gases = Heat gained by steam

$$\therefore (m_a + m_{f_1} + m_{f_2}) \times C_{pg} \times (T_e - T_f) = m_s (h_1 - h_4)$$

$$\therefore (416.67 + 5.99 + 3.67) \times 1 \times (700 - 200) = m_s (3433.7 - 163.4)$$

$$\therefore m_s = 65.18 \text{ kg/s}$$

Power developed by steam turbine plant

$$P_{ST} = W_{ST} - W_P$$

$$= W_{ST} = m_s (h_1 - h_2) \quad \text{(neglecting pump work)}$$

$$= 65.18 \times (3433.7 - 2172.9)$$

$$= \mathbf{82181.58\ kW} \qquad \textit{Ans}$$

Total power produced by combined plant

$$P = P_{GT} + P_{ST} = 111530.86 + 82181.58 = 193712.44 \text{ kW}$$

$$\text{Specific fuel consumption} = \frac{m_f}{P} = \frac{34.77 \times 10^3}{193712.44} = \mathbf{0.179\ kg/kWh} \qquad \textit{Ans}$$

Thermal efficiency of steam cycle

$$\eta_{th(GT)} = \frac{P_{ST}}{\text{Heat gained by steam or heat rejected by flue gases}}$$

$$= \frac{P_{ST}}{m_s (h_1 - h_4)} = \frac{82181.58}{65.18 \times (3433.7 - 163.4)} \times 100 = \mathbf{38.55\%} \qquad \textit{Ans}$$

Thermal efficiency of combined cycle

$$\eta_o = \frac{\text{Net power output}}{\text{Total heat supplied}} = \frac{P_{GT} + P_{ST}}{(m_{f_1} + m_{f_2}) \times CV}$$

$$= \frac{(111530.86 + 82181.58)}{(5.99 + 3.67) \times 41{,}000} \times 100 = \mathbf{48.91\%} \qquad \textit{Ans}$$

Problem 9.12 *In a combined cycle power plant, the air is supplied at a rate of 2000 tones/hr and at temperature 20°C. The pressure ratio is 7 : 1. Inlet pressure to compressor and outlet pressure from the turbine may be taken as 1 bar. The temperature in the gas*

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turbine cycle is limited to 1000°C. The isentropic efficiency for compressor = 80% and for turbine = 85%. The C.V. of oil used = 45,000 kJ/kg.

Data for steam turbine cycle :

The temperature of gas used for steam generation is increased to 1200°C by burning the fuel in the exhaust coming out from gas turbine. The condition of steam generated in the boiler is 50 bar and 500°C. The condenser pressure is 0.1 bar. The temperature of gas going to stack is 200°C.

Find out the following :

(i) Total power generating capacity of the plant, (ii) overall efficiency of the plant.

Take C_{pa} = 1 kJ/kg K, γ = 1.4 for air and C_{pg} = 1.1 kJ/kg K, γ = 1.33 for gas. **[G.U., Dec. 2003]**

Solution : Given Data : m_a = 2000 tones/hr = 555.55 kg/s, T_a = 20°C = 293 K, p_b / p_a = 7, p_a = 1 bar, T_C = 1000°C = 1273 K, η_c = 0.8, η_t = 0.85, CV = 45,000 kJ/kg, T_e = 1200°C = 1473 K, p_1 = 50 bar, T_1 = 500°C, p_2 = 0.1 bar, T_f = 200°C = 473 K, C_{pa} = 1 kJ/kg K, γ = 1.4 for air, C_{pg} = 1.1 kg/kg K, γ = 1.33 for gas, P = ?, η_o = ?

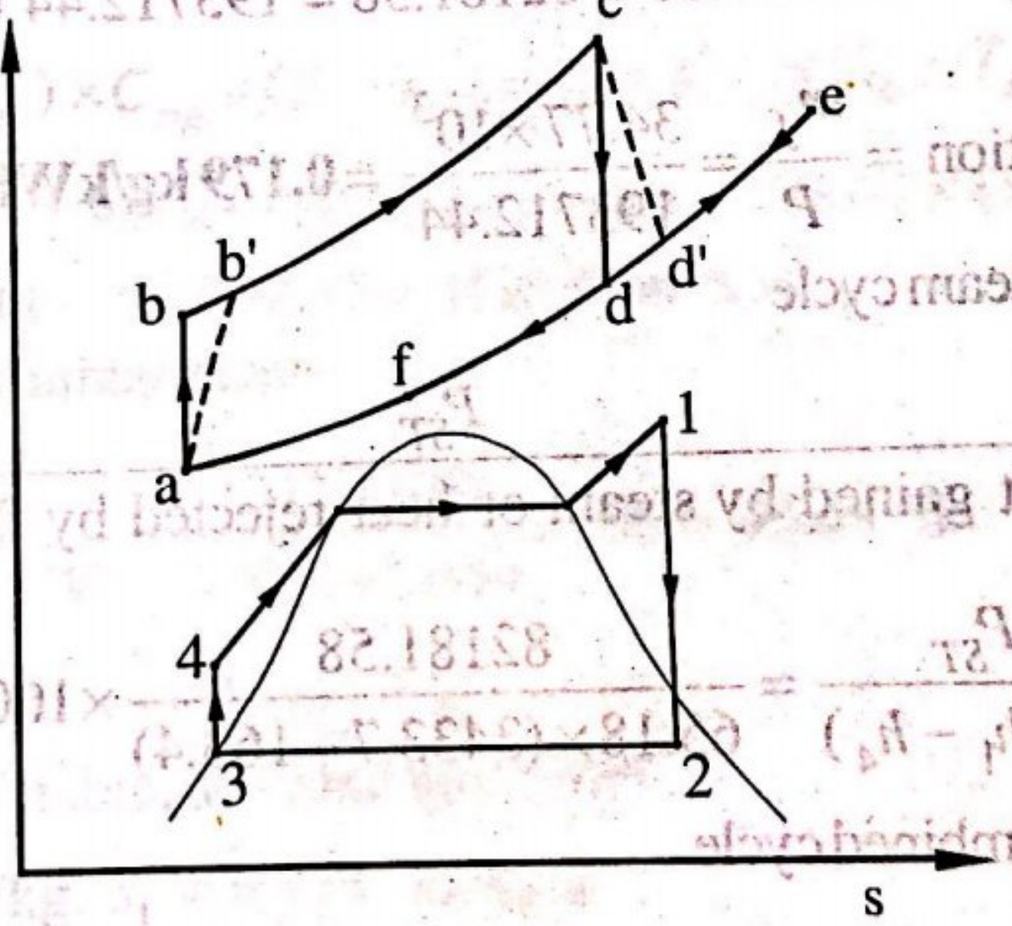


Fig. 9.40

Gas turbine cycle :

For process a-b, $\frac{T_b}{T_a} = \left(\frac{p_b}{p_a}\right)^{\frac{\gamma-1}{\gamma}}$ $\therefore T_b = 293 \times (7)^{\frac{1.4-1}{1.4}} = 510.88\ K$

Also, $\eta_c = 0.8 = \frac{T_b - T_a}{T_b' - T_a}$ $\therefore T_b' = \frac{(510.88 - 293)}{0.8} + 293 = 565.35\ K$

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For process c-d, $\frac{T_c}{T_d} = \left(\frac{p_b}{p_a}\right)^{\frac{\gamma-1}{\gamma}}$ $\therefore T_d = 1273 \times \left(\frac{1}{7}\right)^{\frac{1.33-1}{1.33}} = 785.49\ K$

Also, $\eta_t = 0.85 = \frac{T_c - T_d'}{T_c - T_d}$ $\therefore T_d' = 1273 - 0.85(1273 - 785.49) = 858.61\ K$

Heat balance in combustion chamber

$$m_{f_1} \cdot CV = (m_a + m_{f_1}) \cdot C_{pg} \cdot (T_c - T_b')$$

$$\therefore\ m_{f_1} \times 45000 = (555.55 + m_{f_1}) \times 1.1 \times (1273 - 565.49)$$

$$\therefore\ m_{f_1} = 9.77\ \text{kg/s}.$$

Power produced by gas turbine plant

$$P_{GT} = (m_a + m_{f_1}) \cdot C_{pg} \cdot (T_c - T_d') - m_a \cdot C_{pa}(T_b' - T_a)$$

$$= (555.55 + 9.77) \times 1.1 \times (1273 - 858.61) - 555.55 \times 1 \times (565.35 - 293)$$

$$= 106385.2\ \text{kW} = 106.38\ \text{MW}$$

Heat balance in secondary combustion chamber

$$m_{f_2} \cdot CV = (m_a + m_{f_1} + m_{f_2}) \cdot C_{pg} \cdot (T_e - T_d')$$

$$\therefore m_{f_2} \times 45000 = (555.55 + 9.77 + m_{f_2}) \times 1.1 \times (1473 - 858.61)$$

$$\therefore m_{f_2} = 8.62\ \text{kg/s}$$

Steam turbine cycle :

At pressure 50 bar and 500°C, from superheated steam table

$$h_1 = 3433.7\ \text{kJ/kg}, \quad s_1 = 6.977\ \text{kJ/kg K}$$

At pressure 0.1 bar, from saturated steam table,

$$h_{f_2} = 191.8\ \text{kJ/kg}, \ h_{fg_2} = 2392.8\ \text{kJ/kg},$$

$$s_{f_2} = 0.649\ \text{kJ/kg.K}, \ s_{fg_2} = 7.501\ \text{kJ/kg K}$$

For isentropic process 1-2, $s_1 = s_2$, $\therefore s_1 = s_{f_2} + x_2 s_{fg_2}$

$$\therefore 6.977 = 0.649 + x_2(7.501) \qquad \therefore x_2 = 0.84$$

Now, $h_2 = h_{f_2} + x_2 h_{fg_2} = 191.8 + 0.84(2392.8) = 2210.4\ \text{kJ/kg}$

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Neglecting pump work, $h_4 = h_3 = h_{f_2} = 191.8\ kJ/kg$

Heat balance in HRSG

$$(m_a + m_{f_1} + m_{f_2}) \cdot C_{pg} \cdot (T_e - T_f) = m_s (h_1 - h_4)$$

$$\therefore (555.55 + 9.77 + 8.62) \times 1.1 \times (1473 - 473) = m_s (3433.7 - 191.8)$$

$$\therefore m_s = 194.74\ kg/s$$

Power developed by steam cycle

$$P_{st} = W_{ST} - W_P = m_s (h_1 - h_2) \qquad \text{(Neglecting pump work)}$$

$$= 194.74 \times (3433.7 - 2210.4)$$

$$= 238227.85\ \text{kW} = 238.23\ \text{MW}$$

Total power generated by combined cycle

$$P = P_{GT} + P_{ST} = 106.38 + 238.23 = \mathbf{344.61\ MW} \qquad \textit{Ans}$$

Overall thermal efficiency of plant

$$\eta_o = \frac{P}{\dot{Q}_s} = \frac{P_{GT} + P_{GT}}{(m_{f_1} + m_{f_2}) \cdot CV} = \frac{344.61 \times 10^3}{(9.77 + 8.62) \times 45000} \times 100 = \mathbf{41.64\%} \qquad \textit{Ans}$$

Problem 9.13 *In a gas and steam turbine combined cycle the maximum temperature in the cycle limited to 1100°C. The inlet temperature of air to compressor is 20°C and pressure ratio is 8. The amount of air supplied to gas turbine is 5 times of the theoretical air. In a supplementary firing, the gas temperature is raised to 1000°C before entering into heat recovery boiler. The gas leaves the steam generator at 300°C. The steam is generated at 80 bar and 600°C from the feed water supplied at 150°C. Calculate fuel added in the supplementary firing per kg of air and the mass ratio of air flow to steam flow. Also find out power generating capacity of the plant if the air flow through compressor is 1.5 kg/s. Find efficiency of the plant.*

Take condenser pressure = 0.05 bar, fuel used is methane (CH_4) of fuel = 61,600 kJ/kg, C_p (for gas and air) = 1 kJ/kg K, $\gamma = 1.4$.

Isentropic efficiency of compressor and turbine 100%.

Assume proportion of O_2 in air as 23% by weight. **[G.U. Dec. 2002]**

Solution : Given data : $T_c = 1100°C = 1373\ K$, $T_a = 20°C = 293\ K$, $p_b / p_a = 8$, m_a = (actual) $= 5 \times m_a$ (theoretical), $T_e = 1000°C = 1273\ K$; $T_f = 300°C = 573\ K$, $p_1 = 80\ bar$, $T_1 = 600°C$, $T_4 = 150°C$, $m_{f_2} = ?$, $m_a / m_s = ?$, $P = ?$, $m_a = 1.5\ kg/s$, $\eta_o = ?$, $p_2 = 0.05$ bar, CV of fuel = 61,600 kJ/kg, $C_{pa} = C_{pg} = 1\ kJ/kg\ K$, $\gamma = 1.4$, $\eta_c = \eta_t = 100\%$.

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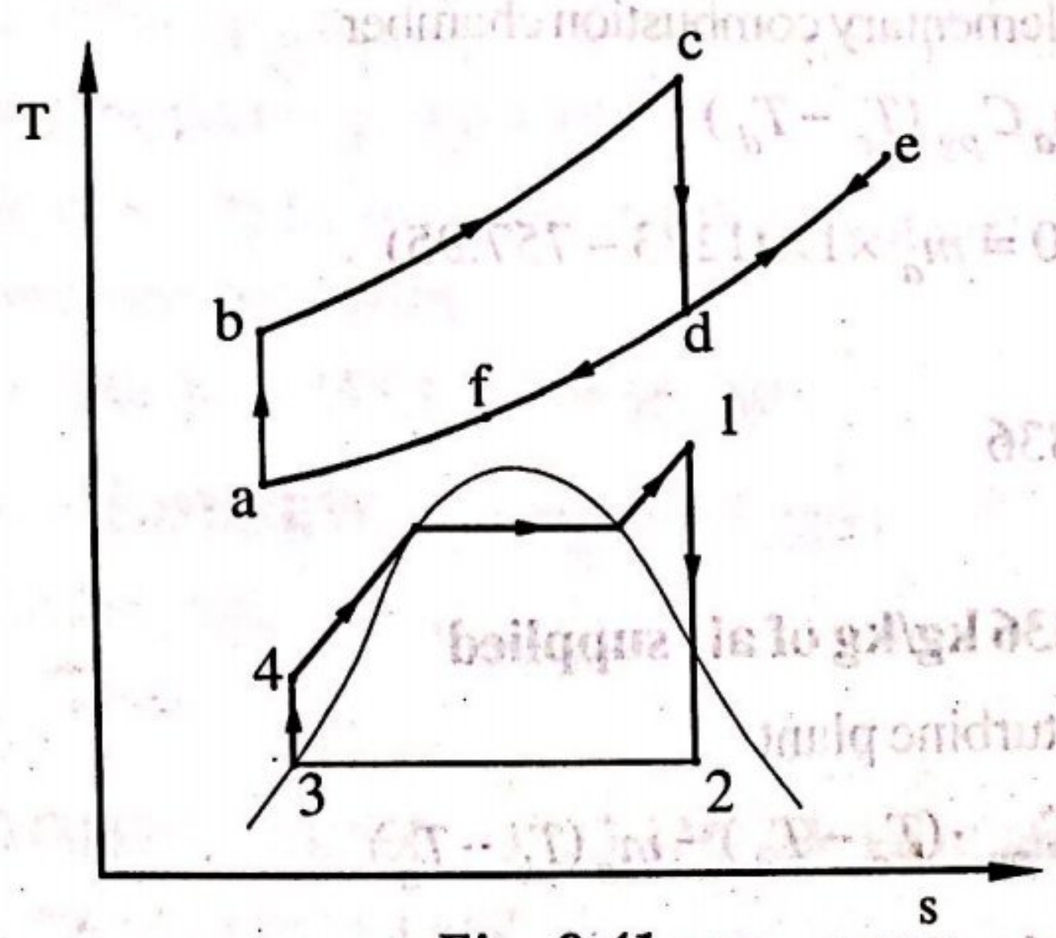


Fig. 9.41

Gas turbine cycle :

For process a-b, $\frac{T_b}{T_a} = \left(\frac{p_b}{p_a}\right)^{\frac{\gamma-1}{\gamma}}$ $\therefore T_b = 293 \times (8)^{\frac{1.4-1}{1.4}} = 530.75\,K$

For process c-d, $\frac{T_c}{T_d} = \left(\frac{p_b}{p_a}\right)^{\frac{\gamma-1}{\gamma}}$ $\therefore T_d = 1373 \times \left(\frac{1}{8}\right)^{\frac{1.4-1}{1.4}} = 757.95\,K$

For combustion process b-c,

The combustion reaction taking place in combustion chamber is given by

$$CH_4 + 2O_2 = CO_2 + 2H_2O$$

$\therefore$ (12 + 4) kg of CH_4 + (2 × 32) kg of O_2
= (12 + 32) kg CO_2 + 2 × (2 + 16) kg of H_2O

$\therefore$ 16 kg of CH_4 + 64 kg of O_2 = 44 kg of CO_2 + 36 kg of H_2O

$\therefore$ 1 kg of CH_4 + 4 kg of O_2 = 2.75 kg of CO_2 + 2.25 kg of H_2O

$\therefore$ Amount of O_2 required = 4 kg/kg of CH_4

$\therefore$ Amount of air required $= \frac{100}{23} \times 4 = 17.39$ kg per kg of CH_4

$\therefore$ Actual air supplied = 5 × theoretical air required

$\therefore m_a = 5 \times 17.39 = 86.95$ kg/kg of fuel

Amount of fuel supplied/kg of air flow in CC-I,

$$m_{f_1} = \frac{1}{86.95} = 0.0115 \text{ kg}$$

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Energy balance in supplementary combustion chamber

$$m_{f_2} \times CV = m_a C_{pg} (T_e - T_d)$$

$$\therefore m_{f_2} \times 61,600 = m_a \times 1 \times (1273 - 757.95)$$

$$\therefore \frac{m_{f_2}}{m_a} = 0.00836$$ **Ans**

$$\therefore m_{f_2} = \textbf{0.00836 kg/kg of air supplied}$$

Power produced by gas turbine plant

$$P_{GT} = (m_a + m_{f_1}) \cdot C_{pg} \cdot (T_c - T_d) - m_a (T_b - T_a)$$

$$= (1 + 0.0115) \times 1 \times (1373 - 757.95) - 1 \times (530.75 - 293)$$

$$= 384.37 \text{ kW/kg of air}$$

Steam turbine cycle :

At 80 bar pressure and 600°C, from superheated steam table

$$h_1 = 3639.5 \text{ kJ/kg}, \; s_1 = 7.019 \text{ kJ/kg K}$$

Enthalpy of feed water $h_4 = C_{pw} T_4 = 4.187 \times 150 = 628.05$ kJ/kg

At 0.05 bar pressure from saturated steam table

$$h_{f_2} = 137.8 \text{ kJ/kg}, \; h_{fg_2} = 2423.7 \text{ kJ/kg}$$

$$s_{f_2} = 0.476 \text{ kJ/kg K}, \; s_{fg_2} = 7.919 \text{ kJ/kg K}$$

For isentropic process 1-2, $s_1 = s_2$ $\quad \therefore s_1 = s_{f_2} + x_2 s_{fg_2}$

$$\therefore 7.019 = 0.476 + x_2 (7.919) \quad \therefore x_2 = 0.826$$

$$\therefore h_2 = h_{f_2} + x_2 h_{fg_2} = 137.8 + 0.826(2423.7) = 2140.36 \, kJ/kg$$

Energy balance in HRSG

$$(m_a + m_{f_1} + m_{f_2}) \cdot C_{pg} \cdot (T_e - T_f) = m_s (h_1 - h_4)$$

$$\therefore (1 + 0.0115 + 0.00836) \times 1 \times (1000 - 300) = m_s \times (3639.5 - 628.5)$$

$$\therefore m_s = 0.237 \text{ kg/kg of air}$$

∴ Mass ratio of air flow to steam flow

$$\frac{m_a}{m_s} = \frac{1}{0.237} = \textbf{4.218 kg of air/kg of steam generated}$$ **Ans**

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Power produced by steam turbine plant

$$P_{ST} = m_s (h_1 - h_2), \text{ neglecting pump work}$$

$$= 0.237 \times (3639.5 - 2140.36) = 355.3 \text{ kW / kg of air}$$

Generating capacity of combined cycle plant

$$P = P_{GT} + P_{ST} = 384.37 + 355.3 = 739.66 \text{ kW}$$

$$= 1.5 \times 739.66 = \mathbf{1109.5\ kW} \quad (\because\ m_a = 1.5 \text{ kg/s})$$

Now, total heat supplied to the plant **Ans**

$$Q_s = (m_{f_1} + m_{f_2}) \times CV$$

$$= (0.0115 + 0.00836) \times 61,600 = 1223.376 \text{ kW/kg of air}$$

$$= 1.5 \times 1223.376 = 1835.064 \text{ kW}$$

Overall combined cycle plant efficiency

$$\eta_o = \frac{P}{Q_s} = \frac{1109.5}{1835.064} \times 100 = \mathbf{60.46\%}$$

Ans

Problem 9.14 *A combined power plant consists of a gas turbine unit and a steam turbine unit, the exhaust gases from the gas turbine is supplied to the steam generator. The pressure ratio for the gas turbine cycle is 8 and the inlet air temperature to the compressor is 15°C, and maximum cycle temperature for gas turbine is 800°C. The temperature of gases leaving the steam generator is 160°C. The condition of steam at steam turbine inlet is 20 bar and 400°C and the condenser pressure is 0.05 bar. The total power output of combined plant is 50 MW. Isentropic efficiencies of air compressor, gas turbine and steam turbine are 80%, 82% and 80% respectively.*

Take $C_p = 1.11$ *kJ/kg K and* $\gamma = 1.33$ *for combustion gases and*

$C_p = 1.005$ *kJ/kg K and* $\gamma = 1.4$ *for air.*

Draw the schematic flow diagram and neglecting the effect of the mass flow rate of fuel, feed pump work and pressure losses, determine :

(i) gas turbine cycle efficiency,

(ii) steam turbine cycle efficiency if the heat supplied in the generator were supplied by an external fuel supply,

(iii) mass flow rates of air and steam, and

(iv) overall efficiency of the combined cycle. **[G.U., Dec 2001]**

Solution : Given data : $p_b / p_a = 8$, $T_a = 15°C = 288$ K, $T_c = 800°C = 1073$ K,

$T_f = 160°C = 433$ K, $p_1 = 20$ bar, $T_1 = 400°C$, $p_2 = 0.05$ bar, $P = 50 \times 10^3$ kW,

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$\eta_c = 0.8$, $\eta_{GT} = 0.82$, $\eta_{ST} = 0.8$, $C_{pa} = 1.005$ kJ/kg K, $\gamma = 1.4$ for air, $C_{pg} = 1.11$ kJ/kg K, $\gamma = 1.33$ for gases, $\eta_{th(GT-plant)} = ?$, $\eta_{th(ST-plant)} = ?$, $m_a = ?$, $m_s = ?$, $\eta_o = ?$

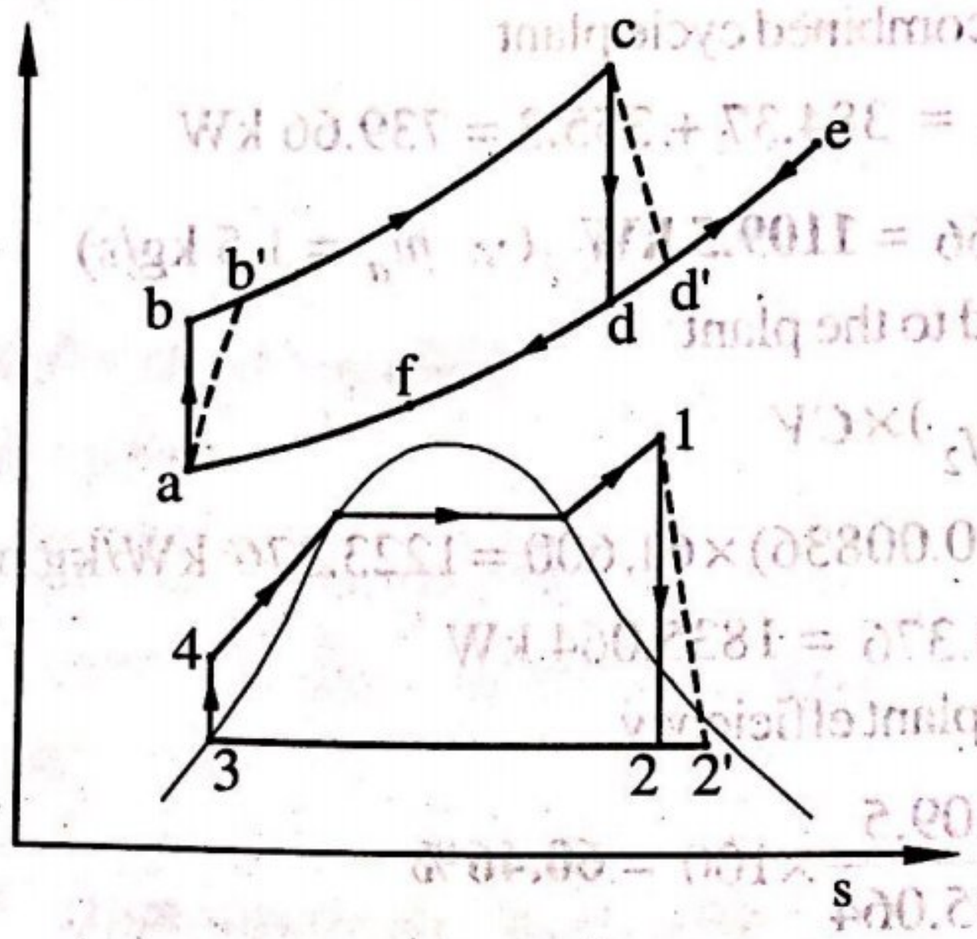


Fig. 9.42

Gas turbine cycle :

For process a-b, $\frac{T_b}{T_a} = \left(\frac{p_b}{p_a}\right)^{\frac{\gamma-1}{\gamma}}$ $\therefore T_b = 288 \times (8)^{\frac{1.4-1}{1.4}} = 521.69\,K$

Also, $\eta_c = 0.8 = \frac{T_b - T_a}{T_b' - T_a}$ $\therefore T_b' = 288 + \frac{(521.69 - 288)}{0.8} = 580.11\,K$

For process c-d, $\frac{T_c}{T_d} = \left(\frac{p_b}{p_a}\right)^{\frac{\gamma-1}{\gamma}}$ $\therefore T_d = 1073 \times \left(\frac{1}{8}\right)^{\frac{1.33-1}{1.33}} = 638.01\,K$

Also, $\eta_{GT} = 0.82 = \frac{T_c - T_d'}{T_c - T_d}$ $\therefore T_d' = 1073 - 0.82(1073 - 638.01) = 716.31K$

Energy balance in combustion chamber-I

$$m_{f_1} \times CV = (m_a + m_{f_1}) \cdot C_{pg} \cdot (T_c - T_b')$$

$$\therefore m_{f_1} \times CV = m_a \cdot C_{pg} \cdot (T_c - T_b') \quad (\because \text{neglecting effect of mass of fuel})$$

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$$\therefore m_{f_1} = 1.11 \times (1073 - 580.11) \times \frac{m_a}{CV}$$

$$\therefore m_{f_1} = 547.12 \times \frac{m_a}{CV} \quad \ldots(i)$$

Power produced by gas turbine plant

$$P_{GT} = (m_a + m_{f_1}) \cdot C_{pg} \cdot (T_c - T_d') - m_a \cdot C_{pa} \cdot (T_b' - T_a)$$

$$= m_a \cdot C_{pg} \cdot (T_c - T_d') - m_a \cdot C_{pa} \cdot (T_b' - T_a)$$

($\because$ neglecting effect of mass of fuel)

$$= [1.11 \times (1073 - 716.31) - 1.005 \times (580.11 - 288)] \cdot m_a$$

$$\therefore P_{GT} = 102.35\ m_a \quad \ldots(ii)$$

Steam turbine cycle :

At pressure 20 bar and 400°C, from superheated steam table

$h_1 = 3248.7$ kJ/kg, $s_1 = 7.130$ kJ/kg K

At pressure 0.05 bar, from saturated steam table

$h_{f_2} = 137.8$ kJ/kg, $h_{fg_2} = 2423.7$ kJ/kg

$s_{f_2} = 0.476$ kJ/kg K, $s_{fg_2} = 7.919$ kJ/kg K

For isentropic process 1-2, $s_1 = s_2$, $\therefore s_1 = s_{f_2} + x_2 s_{fg_2}$

$\therefore 7.130 = 0.476 + x_2 (7.919)$ $\therefore x_2 = 0.84$

$$\therefore h_2 = h_{f_2} + x_2 h_{fg_2} = 137.8 + 0.84(2423.7) = 2174.33\ kJ/kg$$

Neglecting pump work, $h_4 = h_3 = h_{f_2} = 137.8$ kJ/kg

Energy balance in CC-II

$$m_{f_2} \cdot CV = (m_a + m_{f_1} + m_{f_2}) \cdot C_{pg} \cdot (T_e - T_d')$$

Neglecting effect of mass of fuel, we get

$$m_{f_2} \cdot CV = m_a \cdot C_{pg} \cdot (T_e - T_d')$$

$$\therefore m_{f_2} \cdot CV = m_a \times 1.11 \times (1073 - 716.31)$$

($\because T_e = T_c = 800^\circ = 1083$ K, *a*ssumed)

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$$\therefore m_{f_2} = 395.92 \times \frac{m_a}{CV} \quad \text{...(iii)}$$

Total mass of fuel supplied

$$m_f = m_{f_1} + m_{f_2} = 547.12\frac{m_a}{CV} + 395.92\frac{m_a}{CV}$$

$$\therefore m_f = 943.05\frac{m_a}{CV} \quad \text{...(iv)}$$

Energy balance for HRSG

$$(m_a + m_{f_1} + m_{f_2}) \cdot C_{pg} \cdot (T_e - T_f) = m_s (h_1 - h_4)$$

Neglecting effect of fuel, we get

$$m_a \cdot C_{pg} \cdot (T_e - T_f) = m_s (h_1 - h_4)$$

$$\therefore m_a \times 1.11 \times (800 - 160) = m_s (3248.7 - 137.8)$$

$$\therefore m_s = 0.228\, m_a \quad \text{...(v)}$$

Now, steam turbine efficiency is given by

$$\eta_{ST} = \frac{h_1 - h_2'}{h_1 - h_2} \qquad \therefore h_1 - h_2' = (h_1 - h_2) \times \eta_{st}$$

Power produced by steam turbine plant

$$P_{ST} = m_s (h_1 - h_2') = m_s (h_1 - h_2) \times \eta_{ST} \quad (\because \text{neglecting pump work})$$

$$= 0.228 m_a (3248.7 - 2174.33) \times 0.8$$

$$\therefore P_{ST} = 196.27\, m_a \quad \text{...(vi)}$$

Total power produced by combined cycle plant

$$P = P_{GT} + P_{ST} = 102.35 m_a + 196.27 m_a$$

$$\therefore 50 \times 10^3 = 298.62\, m_a$$

$$\therefore m_a = \mathbf{167.43\ kg/s} \quad \textbf{Ans}$$

Gas turbine plant efficiency

$$\eta_{th(GT-plant)} = \frac{P_{GT}}{m_{f_1} \times CV} = \frac{102.35\, m_a}{547.12 m_a} \times 100 = \mathbf{18.7\ \%} \quad \textbf{Ans}$$

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Steam turbine plant efficiency

$$\eta_{th(ST-plant)} = \frac{P_{ST}}{(m_a + m_{f_1} + m_{f_2}) \cdot C_{pg} \cdot (T_e - T_f)}$$

$$= \frac{P_{ST}}{m_a \cdot C_{pg} \cdot (T_e - T_f)} \quad (\because \text{neglecting mass of fuel})$$

$$= \frac{196.27\, m_a}{m_a \times 1.11 \times (800 - 160)} \times 100 = \mathbf{27.63\%} \qquad \textit{Ans}$$

Overall efficiency of combined cycle

$$\eta_o = \frac{P}{(m_{f_1} + m_{f_2}) \cdot CV} = \frac{102.35 m_a + 196.27 m_a}{943.05 \times \frac{m_a}{CV} \times CV} = \mathbf{31.66\ \%} \qquad \textit{Ans}$$

Mass flow rate of steam $m_s = 0.228 m_a = 0.228 \times 167.43 = \mathbf{38.17\ kg/s}$ *Ans*

Problem 9.15 *A combined cycle power plant consists of a gas turbine unit and a steam turbine unit. The exhaust gas from the open cycle gas turbine is supply to the steam generator of the steam turbine cycle at which additional fuel burned in the gas. The pressure ratio for the gas turbine is 8, the air inlet temperature is 15°C, and maximum cycle temperature is 800°C. Combustion in the steam generator raises the inlet temperature of the exhaust gas from gas turbine to 800°C and the gas leaves the steam generator at 100°C. The condition of steam at steam turbine inlet is 60 bar, 600°C and the condenser pressure is 0.05 bar. The isentropic efficiencies of the air compressor, gas turbine and steam turbine are 83%, 85% and 85% respectively.*

Taking $C_p = 1.11$ kJ/kg K, and $\gamma = 1.33$ for the combustion gases, $C_p = 1.005$ kJ/kg K and $\gamma = 1.4$ for the air, neglecting the mass flow rate of fuel, feed pump work and pressure losses.

Calculate : (i) the required flow rates of air and steam for a total power output of 190 MW, (ii) power output of each unit and (iii) overall efficiency of combined plant.

[G.U., Nov. 2000]

Solution : Given data : $p_b / p_a = 8$, $T_a = 15°C = 288$ K, $T_c = 800°C = 1073$ K,

$T_e = 800°C = 1073$ K, $T_f = 100°C = 373$ K, $p_1 = 60$ bar, $T_1 = 600°C$, $p_2 = 0.05$ bar,

$\eta_c = 0.83$, $\eta_{GT} = 0.85$, $\eta_{ST} = 0.85$, $C_{pg} = 1.11$ kJ/kg K, $\gamma = 1.33$ for gas, $C_{pa} = 1.005$ kJ/kg K, $\gamma = 1.4$ for air, P = 190 MW, $m_a = ?$, $m_s = ?$, $P_{ST} = ?$, $P_{GT} = ?$ $\eta_o = ?$

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Gas turbine cycle :

For process a-b, $\frac{T_b}{T_a} = \left(\frac{p_b}{p_a}\right)^{\frac{\gamma-1}{\gamma}}$ $\therefore T_b = 288 \times (8)^{\frac{1.4-1}{1.4}} = 521.69\,\text{K}$

Also, $\eta_c = 0.83 = \frac{T_b - T_a}{T_b' - T_a}$ $\therefore T_b' = 288 + \frac{(521.69 - 288)}{0.83} = 569.56\,\text{K}$

For process c-d, $\frac{T_c}{T_d} = \left(\frac{p_b}{p_a}\right)^{\frac{\gamma-1}{\gamma}}$ $\therefore T_d = 1073 \times \left(\frac{1}{8}\right)^{\frac{1.33-1}{1.33}} = 640.51\text{K}$

Also, $\eta_{GT} = 0.85 = \frac{T_c - T_d'}{T_c - T_d}$ $\therefore T_d' = 1073 - 0.85(1073 - 640.51) = 705.38K$

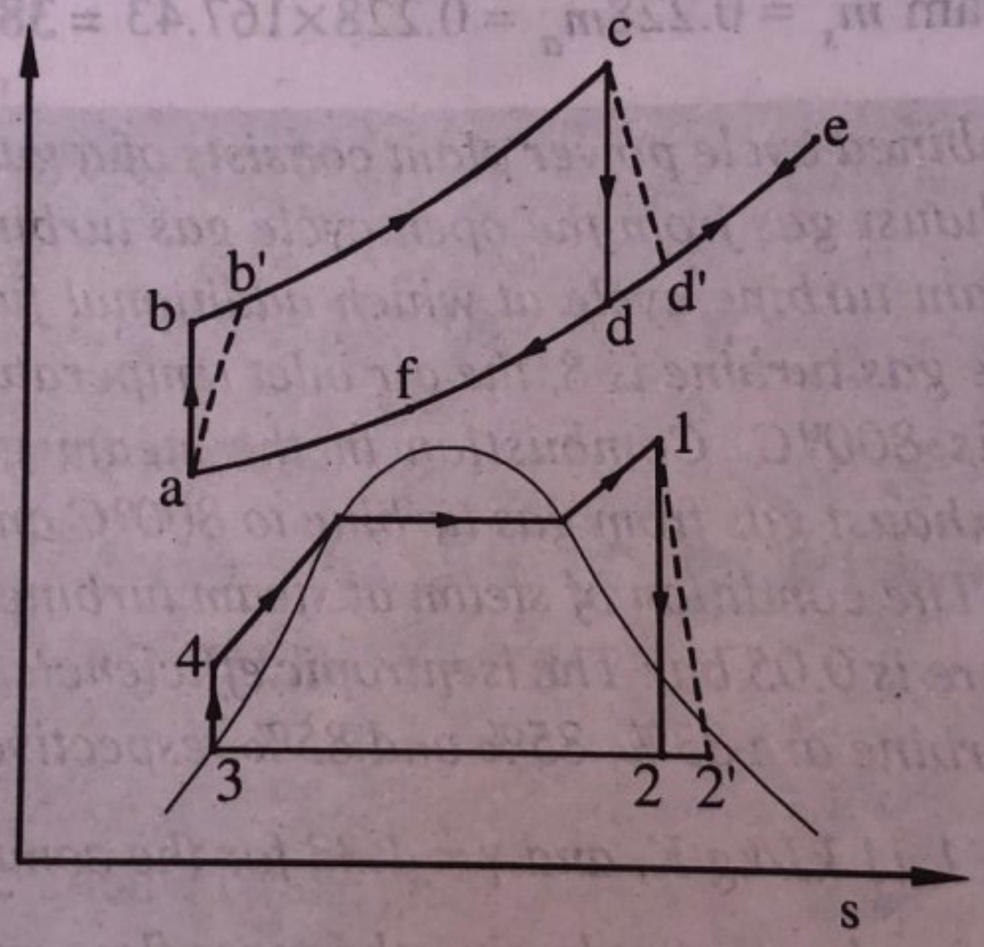


Fig. 9.43

For combustion chamber-I

$$m_{f_1} \times CV = (m_a + m_{f_1}) \cdot C_{pg} \cdot (T_c - T_b')$$

Neglecting mass of fuel

$$m_{f_1} \times CV = m_a \cdot C_{pg} \cdot (T_c - T_b')$$

$$\therefore m_{f_1} \times CV = m_a \times 1.11 \times (1073 - 569.56)$$

$$\therefore m_{f_1} \times CV = 558.82\, m_a \quad \ldots(i)$$

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Power produced by gas turbine plant

$$P_{GT} = (m_a + m_{f_1}) \cdot C_{pg} \cdot (T_c - T_d') - m_a \cdot C_{pa} \cdot (T_b' - T_a)$$

Neglecting mass of fuel

$$P_{GT} = m_a \cdot C_{pg} \cdot (T_c - T_d') - m_a \cdot C_{pa} \cdot (T_b' - T_a)$$

$$= m_a \times 1.11 \times (1073 - 705.58) - m_a \times 1 \times (569.56 - 288)$$

$$= 407.83\, m_a - 281.56\, m_a$$

$$\therefore P_{GT} = 126.27\, m_a \quad \text{...(ii)}$$

Steam turbine cycle :

At pressure 60 bar and 600°C, from superheated steam table

$$h_1 = 3656.2 \text{ kJ/kg}, \; s_1 = 7.166 \text{ kJ/kg K}$$

At pressure 0.05 bar, from saturated steam table

$$h_{f_2} = 137.8 \text{ kJ/kg}, \; h_{fg_2} = 2423.7 \text{ kJ/kg}$$

$$s_{f_2} = 0.476 \text{ kJ/kg K}, \; s_{fg_2} = 7.919 \text{ kJ/kg K}$$

For isentropic process 1-2, $s_1 = s_2$, $\therefore s_1 = s_{f_2} + x_2 s_{fg_2}$

$$\therefore 7.166 = 0.476 + x_2 (7.919) \quad \therefore x_2 = 0.845$$

$$\therefore h_2 = h_{f_2} + x_2 h_{fg_2} = 137.8 + 0.845 \times (2423.7) = 2185.35 \text{ kJ/kg}$$

Also, $h_4 = h_3 = h_{f_2} = 137.8$ kJ/kg (∵ neglecting pump work)

For combustion chamber - II

$$m_{f_2} \cdot CV = (m_a + m_{f_1} + m_{f_2}) \cdot C_{pg} \cdot (T_e - T_d')$$

$$\therefore m_{f_2} \cdot CV = m_a \cdot C_{pg} \cdot (T_e - T_d') \quad (\because \text{neglecting effect of fuel supplied})$$

$$\therefore m_{f_2} \cdot CV = m_a \times 1.11 \times (1073 - 705.38)$$

$$\therefore m_{f_2} \cdot CV = 408.06 m_a \quad \text{...(iii)}$$

Energy balance for HRSG,

$$(m_a + m_{f_1} + m_{f_2}) \cdot C_{pg} \cdot (T_e - T_f) = m_s (h_1 - h_4)$$

Neglecting effect of fuel supplied

$$m_a \cdot C_{pg} \cdot (T_e - T_f) = m_s (h_1 - h_4)$$

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$$\therefore\ m_a \times 1.11 \times (1073 - 373) = m_s (3656.2 - 137.8)$$

$$\therefore\ m_s = 0.221 m_a \quad \text{...(iv)}$$

Power produced by steam turbine plant

$$P_{ST} = m_s \cdot (h_1 - h_2'), (\because \text{neglecting pump work})$$

$$= m_s \cdot (h_1 - h_2) \cdot \eta_{ST}$$

$$= 0.221 \times m_a \times (3656.2 - 2185.35) \times 0.85$$

$$\therefore\ P_{ST} = 276.29\, m_a \quad \text{...(v)}$$

Total power produced by combined cycle

$$P = P_{GT} + P_{ST}$$

$$\therefore 190 \times 10^3 = 126.27 m_a + 276.29 m_a$$

$$\therefore\ m_a = \mathbf{471.98\ kg/s} \quad \textit{Ans}$$

Also, $m_s = 0.221 m_a = 0.221 \times 471.98 = \mathbf{104.3\ kg/s}$ *Ans*

$$P_{GT} = 126.27 m_a = 126.27 \times 471.98 = 59596.9 = \mathbf{59.59\ MW} \quad \textit{Ans}$$

$$P_{ST} = 276.29\, m_a = 276.29 \times 471.98 = 13040.3 = \mathbf{130.41\ MW} \quad \textit{Ans}$$

Overall efficiency of combined cycle power plant

$$\eta_o = \frac{P}{(m_{f_1} + m_{f_2}) \cdot CV} = \frac{190 \times 10^3}{558.82 m_a + 408.06 m_a} \times 100$$

$$= \frac{196.51}{m_a} \times 100 = \frac{196.51}{421.2} \times 100 = \mathbf{46.65\ \%} \quad \textit{Ans}$$

Problem 9.16 *A combined cycle power plant consists of gas turbine unit and steam turbine unit. The inlet condition of air is 1 bar and 20°C. The pressure ratio is 12 and the gas turbine inlet temperature is 1300°C. Fuel used in natural gas having lower calorific value of 42000 kJ/kg. The gas leaves HRSG at 150°C. The steam pressure and temperature are 40 bar and 500°C respectively. The steam at 2 bar is bled from the steam turbine for heating the feed water in the deaerator and the feed water after the deaerator is pumped to the boiler. The condenser pressure is 0.05 bar. The isentropic efficiencies of compressor, gas turbine and steam turbine are 85%, 82% and 82 respectively. The combustion efficiency and HRSG effectiveness are 0.98 and 0.92 respectively. The mechanical efficiency of compressor and turbine each is 99%. The generator efficiency is 98%. The pump efficiency is 90%.*

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Calculate (i) Mass of fuel supplied per kg of air (ii) Gas turbine and steam turbine cycle efficiency, and (iii) Combined cycle efficiency,
Take, C_{pa} = 1.005 kJ/kg K, γ = 1.4, for air and C_{pg} = 1.12 kJ/kg K, γ = 1.3 for gas.

Solution : Given data : $T_a = 20^oC = 293$ K, $p_a = 1$ bar, $p_b / p_a = 12$, $T_c = 1300^oC = 1573$ K, C.V. = 42,000 kJ/kg, $T_f = 150^oC = 423$ K, $p_1 = 40$ bar, $T_1 = 500^oC$, $p_{2a} = 2$ bar, $p_2 = 0.05$ bar, $\eta_c = 0.85$, $\eta_{GT} = 0.82$, $\eta_{ST} = 0.82$, $\eta_{comb} = 0.98$, $\varepsilon_{HRSG} = 0.92$, η_m (compressor, turbine) = 0.99, $\eta_p = 0.9$, m_f = ?, $\eta_{th(GT-plant)}$ = ?, $\eta_{th(ST-plant)}$ = ?, η_o = ?, $C_{pa} = 1.005$ kJ/kg K, $\gamma = 1.4$ for air, $C_{pg} = 1.12$ kJ/kg K, $\gamma = 1.3$ for gas,

Gas turbine cycle :

For process a-b, $\dfrac{T_b}{T_a} = \left(\dfrac{p_b}{p_a}\right)^{\frac{\gamma-1}{\gamma}}$ $\therefore T_b = 293 \times (12)^{\frac{1.4-1}{1.4}} = 595.94\,K$

Also, $\eta_c = 0.85 = \dfrac{T_b - T_a}{T_b' - T_a}$ $\therefore T_b' = 293 + \dfrac{(595.94 - 293)}{0.85} = 649.4\,K$

For process c-d, $\dfrac{T_c}{T_d} = \left(\dfrac{p_b}{p_a}\right)^{\frac{\gamma-1}{\gamma}}$ $\therefore T_d = 1573 \times \left(\dfrac{1}{8}\right)^{\frac{1.3-1}{1.3}} = 973.47\,K$

Also, $\eta_{GT} = 0.82 = \dfrac{T_c - T_d'}{T_c - T_d}$ $\therefore T_d' = 1573 - 0.82(1573 - 973.47) = 1081.38\,K$

Energy balance for combustion chamber of gas turbine

$$m_f \cdot CV \times \eta_{comb} = (m_a + m_f) \cdot C_{pg} \cdot (T_c - T_b')$$

$$\therefore m_f \times 42,000 \times 0.98 = (m_a + m_f) \times 1.12 \times (1573 - 649.4)$$

$$\therefore m_f = 0.02578\, m_a$$

$$\therefore m_f = \mathbf{0.02578\ kg/kg\ of\ air}$$ **Ans**

Heat supplied to GT plant

$$Q_{s_1} = m_f \times CV$$

$$= 0.02578 \times 42,000 = 1082.76\ \text{kJ/kg of air}$$

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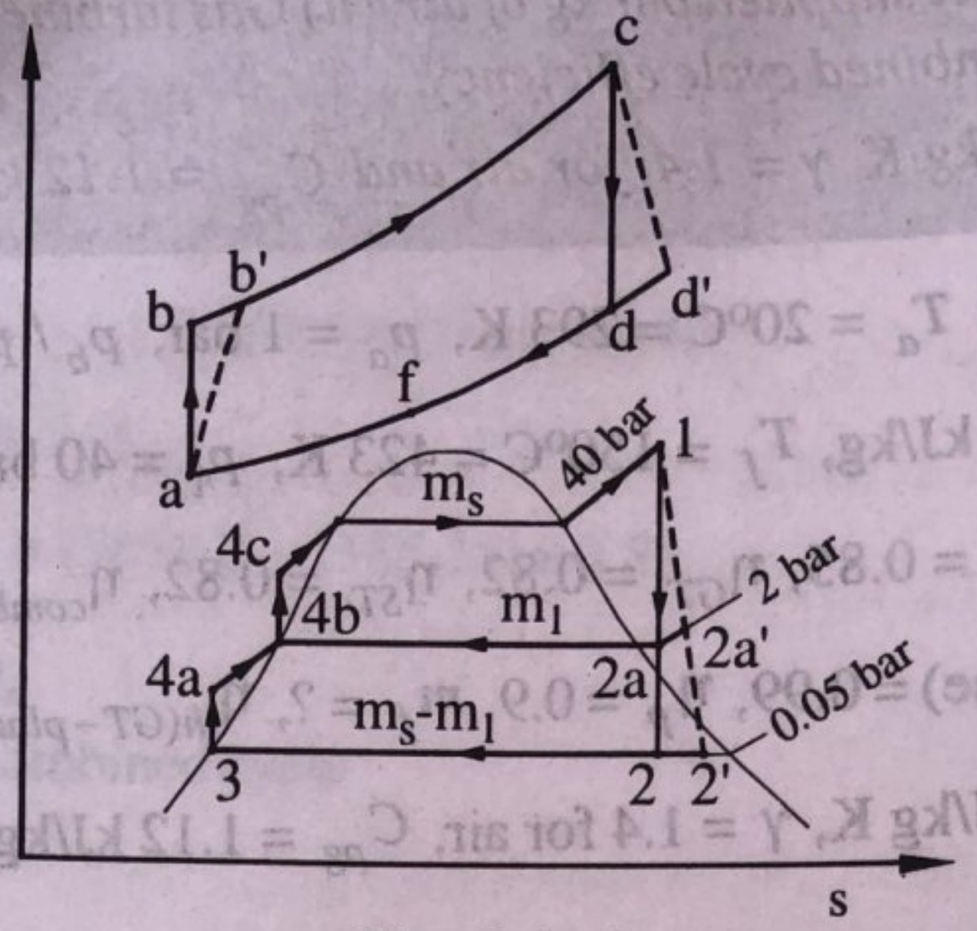


Fig. 9.44

Work produced by gas turbine plant

$$W_{GT} = (m_a + m_f) \cdot C_{pg} \cdot (T_c - T_d') \times \eta_m - m_a \cdot C_{pa} \cdot (T_b' - T_a) \times \frac{1}{\eta_m}$$

$$= (m_a + 0.02578 m_a) \times 1.12 \times (1573 - 1081.38) \times 0.99$$

$$- m_a \times 1.005 \times (649.4 - 293)$$

$$= 197.36 m_a = 197.36 \text{ kJ/kg of air}$$

Gas turbine plant efficiency

$$\eta_{th(GT-plant)} = \frac{W_{GT}}{Q_{s1}} = \frac{197.36}{1082.76} \times 100 = \mathbf{18.22\%} \qquad \textit{Ans}$$

Steam turbine cycle :

At pressure 40 bar and 500°C, from superheated steam table,

$h_1 = 3445.0$ kJ/kg, $s_1 = 7.091$ kJ/kg K

At pressure 2 bar, from saturated steam table

$h_{f2a} = 504.7$ kJ/kg, $h_{fg2a} = 2201.6$ kJ/kg

$s_{f2a} = 1.5201$ kJ/kg K, $s_{fg2a} = 5.5967$ kJ/kg K, $v_{f2a} = 0.001061$ m^3/kg

At pressure 0.05 bar, from saturated steam table

$h_{f2} = 137.8$ kJ/kg, $h_{fg2} = 2423.7$ kJ/kg

$s_{f2} = 0.476$ kJ/kg K, $s_{fg2} = 7.919$ kJ/kg K, $v_{f2} = 0.001005$ m^3/kg

For isentropic process 1-2, $s_1 = s_2$, $s_1 = s_{f_2} + x_2 s_{fg_2}$

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$$\therefore 7.091 = 0.476 + x_2(7.919) \quad \therefore x_2 = 0.835$$

$$\therefore h_2 = h_{f2} + x_2 h_{fg2} = 137.8 + 0.835(2423.7) = 2162.39\,kJ/kg$$

For isentropic process 1-2a, $s_1 = s_{2a}$

$$\therefore 7.091 = 1.5301 + x_{2a}(5.5967) \quad \therefore x_{2a} = 0.993$$

$$\therefore h_{2a} = h_{f2a} + x_{2a} h_{fg2a} = 504.7 + 0.993(2201.6) = 2692.22 \text{ kJ/kg}$$

$h_3 = h_{f2} = 137.8$ kJ/kg, $h_{4b} = h_{f2a} = 504.7$ kJ/kg

Also, $\eta_{ST} = \dfrac{h_1 - h'_{2a}}{h_1 - h_{2a}} = \dfrac{h_1 - h'_2}{h_1 - h_2}$

$$\therefore 0.82 = \frac{3445 - h'_{2a}}{3445 - 2692.22} = \frac{3445 - h'_2}{3445 - 2162.39}$$

$\therefore h'_{2a} = 2827.72$ kJ/kg and $h'_2 = 2393.25$ kJ/kg

Pump work supplied to BFP

$$\therefore W_{p2} = \frac{v_{f2a}(p_1 - p_{2a})}{\eta_p} = \frac{0.001061 \times (40 - 2) \times 10^5}{0.9}$$

$= 4.479$ kJ/kg of steam

But, $W_{p2} = m_s(h_{4c} - h_{4b})$

$$\therefore 4.479 = 1 \times (h_{4c} - 504.7)$$

$$\therefore h_{4c} = 509.18 \text{ kJ/kg of steam}$$

Pump work supplied to CEP

$$W_{p1} = \frac{v_{f2}(p_{2a} - p_2)}{\eta_p} = \frac{0.001005 \times (2 - 0.05) \times 10^5}{0.9}$$

$= 0.2177$ kJ/kg of steam

But $W_{p1} = (h_{4a} - h_3) \times (m_s - m_1)$

$$\therefore 0.2177 = 1 \times (h_{4a} - 137.8)$$

$$\therefore h_{4a} = 138.02 \text{ kJ/kg of steam}$$

Total pump work = 4.479 + 0.2177 = 4.6967 kJ/kg of steam

Energy balance for HRSG

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$$(m_a + m_f) \cdot C_{pg} \cdot (T'_d - T_f) \cdot \varepsilon_{HRSG} = m_s (h_1 - h_{4c})$$

$$\therefore (1 + 0.02578) \times 1.12 \times (1081.38 - 423) \times 0.92 = m_s (3445 - 509.18)$$

$$\therefore m_s = 0.237 \text{ kg/kg of air}$$

Total pump work /kg of air = $4.6967 \times 0.237 = 1.1132$ kJ/kg of air

Energy balance for deaerator

$$m_{s1} h'_{2a} + (m_s - m_{s_1}) h_{4a} = m_s h_{4b}$$

$$\therefore m_{s1} h_{2a} - m_{s_1} h_{4a} = m_s h_{4b} - m_s h_{4a}$$

$$\therefore m_{s1} = m_s \frac{(h_{4b} - h_{4a})}{(h'_{2a} - h_{4a})}$$

$$\therefore m_{s1} = 0.237 \times \left(\frac{504.7 - 138.02}{2827.72 - 138.02} \right) = 0.0323 \text{ kg/kg of air}$$

Work produced by steam turbine

$$W_{ST} = m_s (h_1 - h'_{2a}) + (m_s - m_{s_1})(h'_{2a} - h'_2)$$

$$= 0.237 \times (3445 - 2827.72) + (0.237 - 0.0323) \times (2827.72 - 2393.25)$$

$$= 235.23 \text{ kJ/kg of air}$$

Net work produced by steam power plant

$$W_{ST(net)} = W_{ST} - W_P = 235.23 - 1.1132 = 234.12 \text{ kJ/kg of air}$$

Steam turbine plant efficiency

$$\eta_{th(ST-plant)} = \frac{W_{ST(net)}}{(m_a + m_f) \cdot C_{pg} \cdot (T'_d - T_f)}$$

$$= \frac{234.12}{(1 + 0.02578) \times 1.12 \times (1081.38 - 423)} \times 100 \times (1081.38 + 423)$$

$$= \mathbf{30.95\%}$$ ***Ans***

Total heat supplied to the plant

$$Q_s = Q_{s_1} + Q_{s_2} = 1082.76 + 0 = 1082.76 \text{ kJ/kg of air}$$

Combined cycle efficiency

$$\eta_o = \frac{W_{GT(net)} + W_{ST(net)}}{Q_s} = \frac{197.36 + 234.12}{1082.76} \times 100 = \mathbf{39.85\%}$$ ***Ans***

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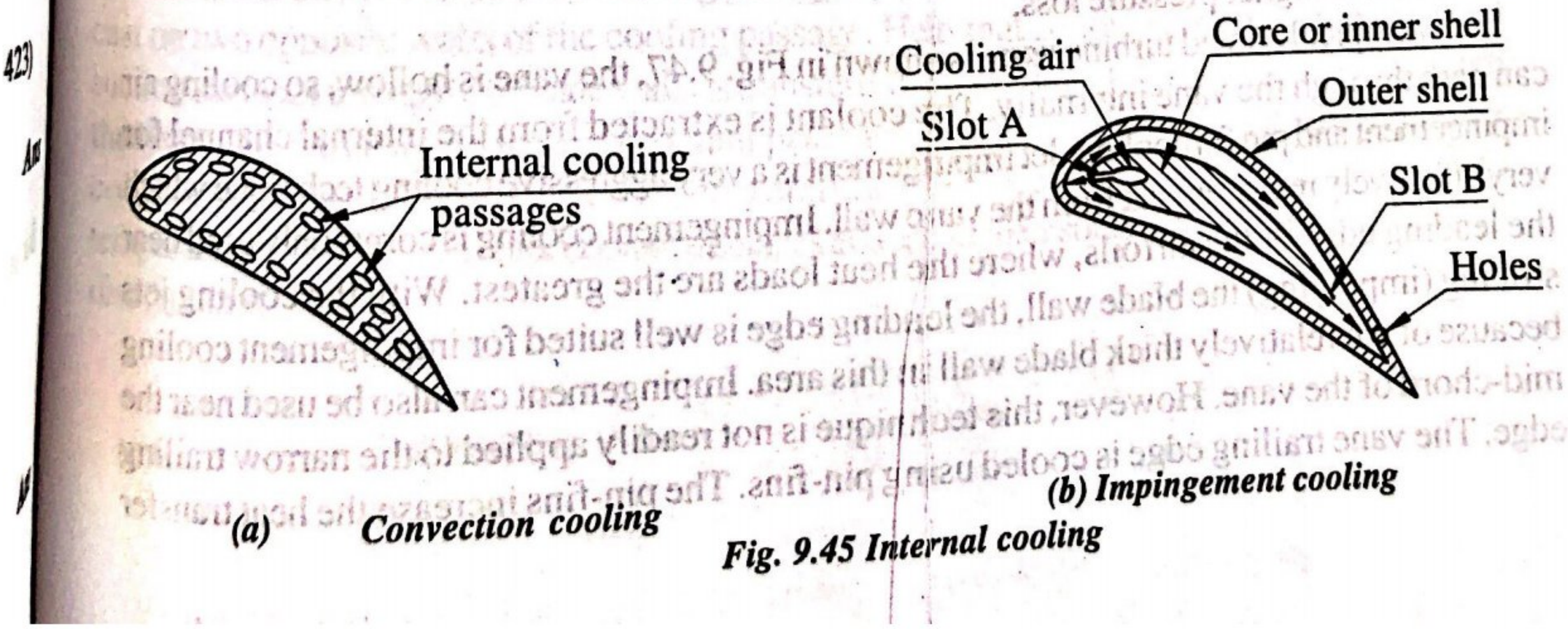
9.25 Gas Turbine blade cooling

The efficiency and power output of a gas turbine increases with higher turbine inlet gas temperature. Modern gas turbine vanes and blades are exposed to gas with temperatures (about 1500°C) which far exceeds the melting point of the component material (metal airfoils). The level and variation in the temperature within the blade material causes thermal stresses which limits the use of high turbine inlet temperature. Therefore, the blades and vanes have to be cooled in order to lower the blade temperature. The objective of blade cooling is to keep the metal temperature at the safe level in order to ensure a long creep life and low oxidation rates. Cooling air around 650°C is extracted from the compressor and passes through the airfoils. With the hot gases and cooling air, the temperature of the blades can be lowered to approximately 1000°C, which is permissible for reliable operation of the turbine. However, the extraction of air from compressor decreases the efficiency of the turbine, since less air is available for power generation. Maximum cooling with minimum cooling air is therefore needed for effective blade cooling.

There are following two types of blade cooling techniques employed to bring down the temperature of the blade material temperature below its melting point as internal blade cooling and external blade cooling.

(1) Internal cooling

In this method of cooling, relatively cold air from the compressor is bypassed and passed through the hollow passages (circular or elliptical) inside the turbine blade as shown in Fig. 9.45(a). In the internal cooling, heat is removed by conduction and convection. Relatively hotter air escapes to the main flow from the blade tips after traversing the entire blade length in the cooling passages. As shown in Fig. 9.45(b) the flow of air impinges in particular manner is called *impingement cooling*. In this case, the cooling air first admitted into the inner shell from which it comes out through narrow and long slots (A and B). The air coming from slot A and slot B impinges the inner surface of the leading and trailing edges respectively. The air from slot A after impinging, flows over the inner shell and comes out by the holes provided on trailing edge.



(a) Convection cooling *(b) Impingement cooling*

Fig. 9.45 Internal cooling

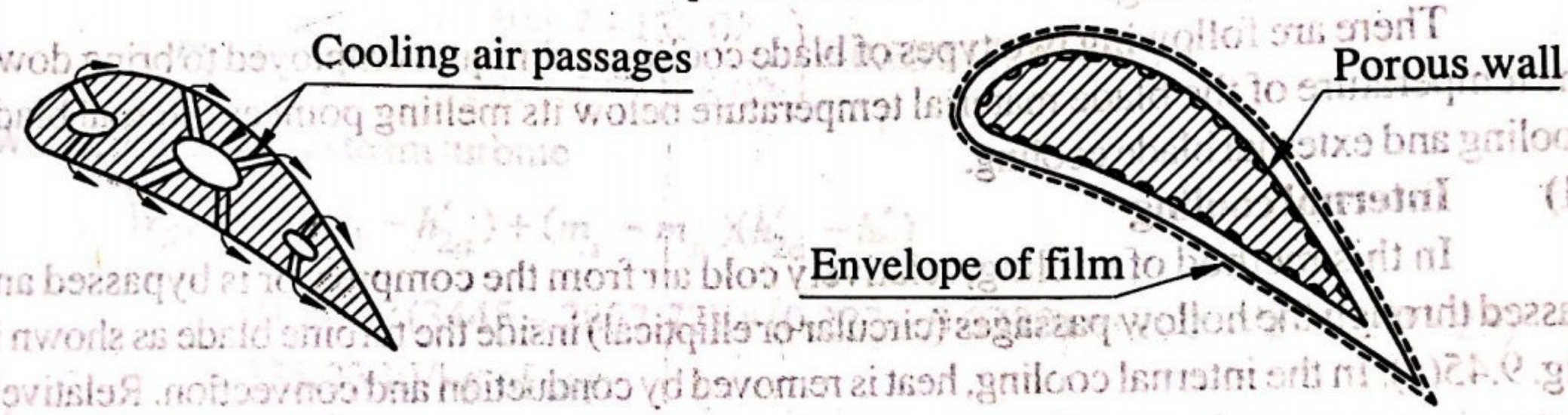
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(2) External cooling

In external cooling, the bypassed air is exited out through small holes at discrete locations of the turbine blade. This relatively cold air creates a protective blanket that protects the turbine blade from the hot gas environment. External cooling can be two types as (a) Film cooling (b) Transpiration or effusion cooling.

Film cooling: In film cooling as shown in Fig. 9.46(a), relatively cold air extracted from the compressor is injected through the drilled holes or precise cast passages on the external blade surface. The cooling air flowing out of these small holes forms a thin film cooling layer over the blade surfaces. This relatively cold air layer protects the turbine blade against the very hot gases at outside of the blades.

Transpiration cooling: In transpiration cooling as shown in Fig. 9.46(b), the relatively cold air extracted from compressor is allowed to effuse from pores of the porous blade metal. This provides blanket of cold air which insulates the turbine blade against the very hot gas at outside of the blades. Effusion of cold air provides uniform cooling of the blade.



(a) Film cooling *(b) Transpiration cooling*

Fig. 9.46 External cooling

A number of cooling methods are applied to different part of the vane or blade. To make the cooling systems more efficient and spend a minimum of air, the cooling systems nowadays usually include features that increase the heat transfer coefficient and/or increasing the heat transfer surface area. The heat transfer coefficient is increased by enhancement of the flow turbulence and by breaking the flow boundary layer. The penalty paid for the increased heat transfer is higher pressure loss.

A typical cooled turbine ***vane*** is shown in Fig. 9.47, the vane is hollow, so cooling air can pass through the vane internally. The coolant is extracted from the internal channel for impingement and pin fin cooling. Jet impingement is a very aggressive cooling technique which very effectively removes heat from the vane wall. Impingement cooling is commonly used near the leading edge of the airfoils, where the heat loads are the greatest. With the cooling jets striking (impinging) the blade wall, the leading edge is well suited for impingement cooling because of the relatively thick blade wall in this area. Impingement can also be used near the mid-chord of the vane. However, this technique is not readily applied to the narrow trailing edge. The vane trailing edge is cooled using pin-fins. The pin-fins increase the heat transfer

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area while effectively mixing the coolant air to lower the wall temperature of the vanes. After impinging on the walls of the airfoil, the coolant exits the vane and provides a protective film on the vane's external surface. Similarly, the coolant traveling through the pin-fin array is ejected from the trailing edge of the airfoil. The trailing edge pin-fin channel normally has ejection holes through which the spent coolant exhausts to the main stream flow.

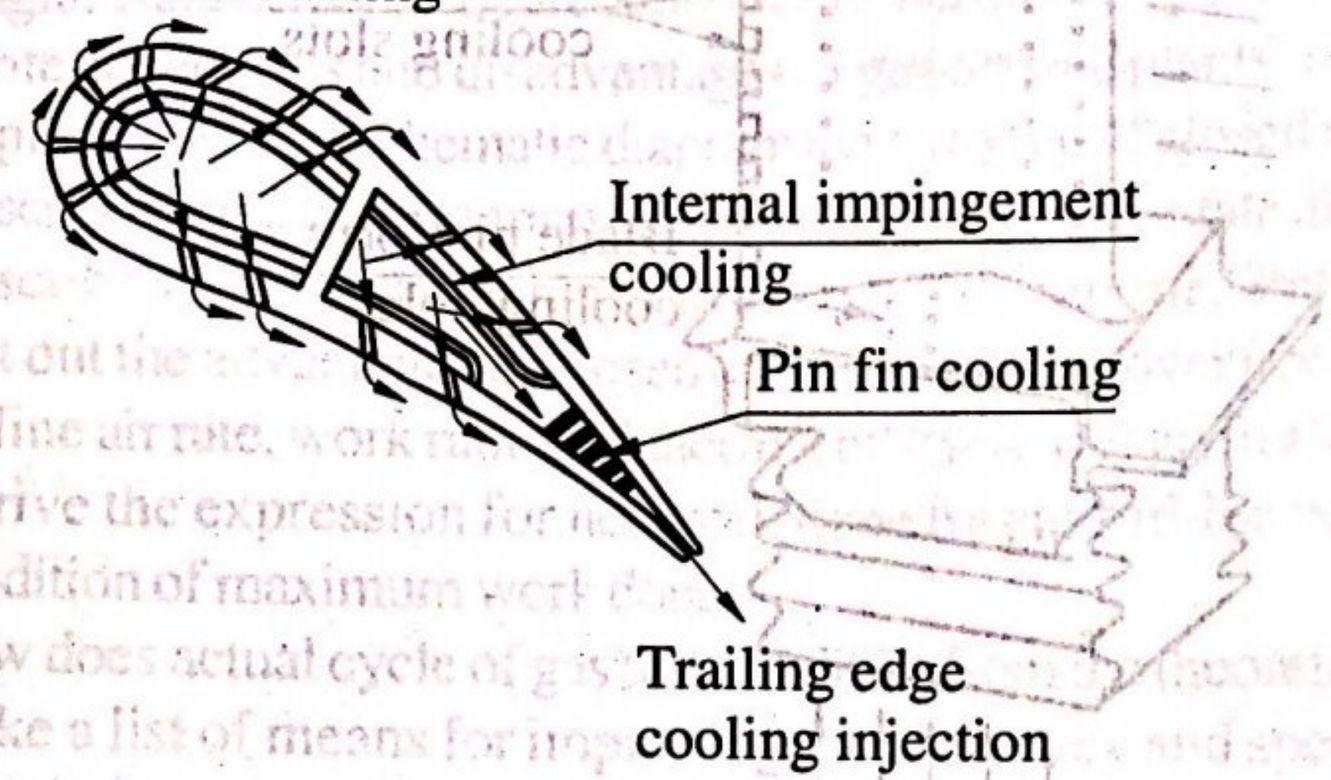


Fig. 9.47 A typical cooled turbine vane cross section with flow of air for cooling

The schematic of a modern gas turbine ***blade*** with common cooling techniques (internal and film cooling) is shown in Fig. 9.48. The leading edge, pressure surfaces and suction surfaces are cooled by film cooling. The leading edge is cooled by jet impingement with film cooling, the middle portion is cooled by serpentine rib-roughened passages with local film cooling, and the trailing edge is cooled by pin fins with trailing edge injection. Although the techniques used to cool the blades are similar to those used to cool the vanes, the heat transfer trends in the vanes and blades are very different. Because the blades are rotating, the flow of the coolant in the passages is altered. Therefore, the effect of rotation on the internal heat transfer enhancement must be considered. Rib turbulators are the most frequently used method to enhance the heat transfer in the internal serpentine cooling passages. The rib turbulence promoters are typically cast on two opposite walls of the cooling passage. Heat that conducts from the pressure and suction surfaces through the blade walls is transferred to the coolant passing internally through the blade. Heat transfer in rotating coolant passages is very different from that in stationary coolant passages. Both Coriolis and rotating buoyancy forces alter the flow and temperature profiles in the rotor coolant passages and affect their surface heat transfer coefficient distributions.

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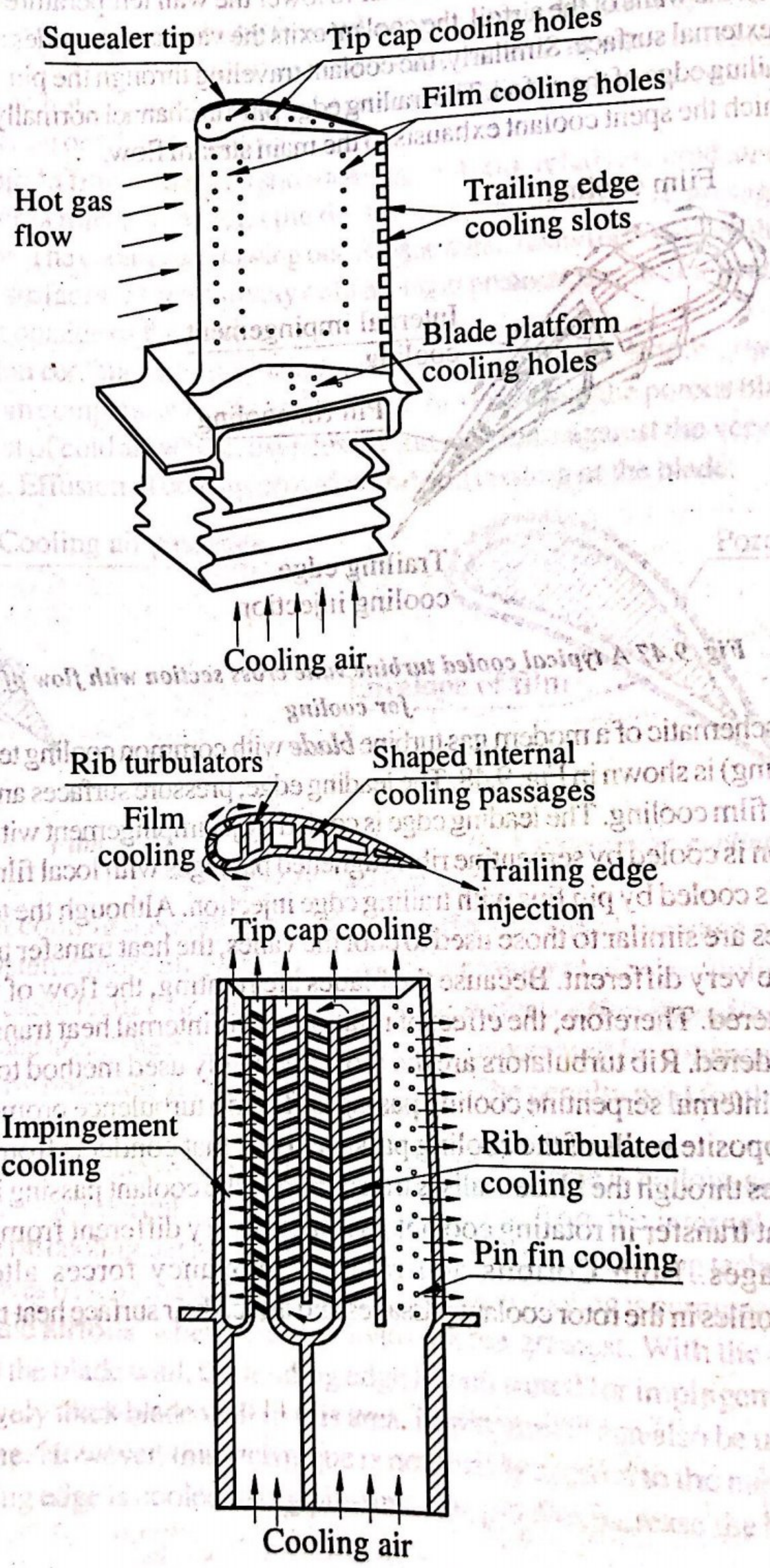


Fig. 9.48 Schematic of a modern gas turbine blade with common cooling techniques

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Exercise

9.1 Give the advantages and limitations of gas turbine power plant.
9.2 What are the major fields of application of gas turbine?
9.3 State the classification of gas turbine power plants.
9.4 Explain with neat sketch the open cycle gas turbine power plant. Also show it on T-S diagram and derive an expression for the thermal efficiency.
9.5 Write advantages and disadvantages of gas turbine plants over steam turbine plants.
9.6 Explain with neat schematic diagram the working of closed cycle gas turbine.
9.7 Describe with a neat diagram closed cycle gas turbine, state also its merrits and demerits.
9.8 Describe with a neat diagram, closed cycle gas turbine. State its applications.
9.9 List out the advantages of closed cycle gas turbine over open cycle gas turbine plant.
9.10 Define air rate, work ratio and thermal efficiency of gas turbine power plant.
9.11 Derive the expression for net work done by gas turbine power plant. Also state the condition of maximum work done.
9.12 How does actual cycle of gas turbine differ from the theoretical?
9.13 Make a list of means for improving the efficiency and specific output of simple gas turbine plant.
9.14 Explain in brief, with rough graphs, the effect of regenerative, simple and complete gas turbine cycle on specific fuel consumption for all values of compression ratio.
9.15 Discuss the effect of operating variables on air rate and work ratio, used in gas turbine.
9.16 Discuss the effect of operating variables on air rate and work ratio.
9.17 Explain water injection for improving power output of gas turbine plant.
9.18 What are the main requirements of a gas turbine combustion chamber ?
9.19 What are the main types of gas turbine combustion chamber ?
9.20 What is primary and secondary air in gas tubine combustion chamber ? What are their function ?
9.21 Explain with the sketch the Can type combustor with swirl flow flame stabilizer.
9.22 Explain with the sketch the Can type combustor with bluff-body flame stabilizer.
9.23 Define combined cycle power plant.
9.24 Discuss the advantages of combined cycle power generation. Why is it so important in the present day energy scenario ?
9.25 State the different arrangement of combined cycle power plants and explain any one with neat sketch.
9.26 Explain with a neat schematic diagram the working of combined cycle. Also state its importance and merits.
9.27 State and briefly explain different arrangement of topping cycle co-generation plants.
9.28 Derive an expression of combined cycle plant efficiency in terms of topping and bottoming cycle efficiencies.
9.29 Describe the arrangement of a combined cycle consisting of simple gas turbine plant

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and waste heat boiler. Write the advantages of a combined cycle power plant.

9.30 Explain how a combined cycle power plant increases the thermal efficiency of the plant.

9.31 Why blade cooling is required in gas turbine?

9.32 Explain with neat sketch the blade cooling of gas turbine.

Unsolved Numerical Problems

9.1 An open cycle gas turbine takes air at 1 bar, and 18°C. The pressure ratio is 4 and compression takes place in two stages with perfect intercooling. The maximum temperature of cycle is 650°C. The air is heated in regenerator before it enters the combustion chamber. The effectiveness of regenerator is 0.7. The isentropic efficiencies of both compressors and turbine is 0.86. Calculate thermal effieicncy of gas turbine plant. Take $\gamma = 1.4$ and $c_p = 1.05$ *kJ/kg K* for both air and gases.

[Ans : 44.9%]

9.2 In an open cycle gas turbine plant air enters at 1 bar and 27°C and compressed to 4.5 bar pressure. The pressure loss in the combustion chamber is 0.05 bar. The combustion effieicney is 92%. The maximum cycle temperature is 600°C. The isentropic efficiencies of compressor and turbine are 0.85 and 0.86 respectively. The calorific value of fuel is 40,000 kJ/kg K. Calculate (i) power generated by plant when mass flow of air is 20 kg/s, (ii) specific fuel consumption. Take $\gamma = 1.4$ and $C_p = 1$ *kJ/kg K*.

[Ans : 1425 kW, 0.214 kg/s]

9.3 In an oil gas turbine installation, air is taken at pressure of 1 bar and 300 K and compressed to a pressure of 4 bar. The oil (C.V. = 43,000 kJ/kg) is burnt in the combustion chamber to raise temperature of air to 873 K. If the air flows at the rate 2 kg/s. Calculate net power generated and air fuel ratio. Take, $\gamma = 1.4$ and $C_p = 1$ *kJ/kg K*.

[Ans : 285.15 kW, 99.65 : 1]

9.4 A gas turbine plant relates following data :

Compressor inlet air condition = 1 bar, 20°C, and 65% relative humidity
Pressure ratio = 6
Isentropic efficiency of compressor = 0.82
Temperature of water which is injected into air after the compression = 52°C
The condition of air after water injection = 95°C and saturated
Mass flow rate of air at compressor inlet = 20 kg/s

Calculate mass flow rate of water injection.

Take, $C_p = 1.005$ kJ/kg K and $\gamma = 1.4$ for air and $C_p = 4.18$ kJ/kg K for water.

[Ans: 1.53 kg/sec]

9.5 A combined cycle power plant consists of gas turbine unit and steam turbine unit. The inlet condition of air is 1 bar and 20°C. The pressure ratio is 12 and gas turbine inlet temperature is 1300°C. Fuel used is natural gas (CV = 42000 kJ/kg). The gas leaves heat recovery steam generator at 150°C. The steam pressure and temperature are 40

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bar and 500°C. The condenser pressure is 0.05 bar. The isentropic efficiencies of compressor, gas turbine, and steam turbine is 100% each. Neglect pump work. Calculate (i) mass of fuel supplied per kg of air supplied, (ii) gas and steam turbine cycle efficiency and (iii) combined cycle efficiency.

Take, $C_{pa} = 1.005$ kJ/kg K, $\gamma = 1.4$ for air, $C_{pg} = 1.12$ kJ/kg K, $\gamma = 1.3$ for gas.

[Ans : (i) 0.27 kg/kg of air, (ii) 34.26%, 38.8%, (iii) 56.1 %]

9.6 In a combined cycle power plant, compressor takes air at 1 bar pressure and 20°C. The maximum temperature of gas turbine is limited to 900°C. The pressure ratio is 8. The exhaust gases coming from gas turbine is heated further to 900°C before entering into HRSG. The steam is generated at 40 bar and 500°C. The exhaust gas leaving the HRSG is at 180°C. Assume isentropic compression in compressor and expansion in both turbines. Neglect pump work.

Take, $C_{pa} = C_{pg} = 1$ kJ/kg K, $\gamma = 1.4$, CV of fuel = 40,000 kJ/kg

Calculate : (i) Power generated by combined cycle power plant if mass of air supplied to compressor is 500 kg/s. (ii) Thermal efficiency of combined cycle

[Ans : (i) 292 MW, (ii) 49.4%]

9.7 In a combined cycle power plant, compressor takes air at 1 bar and 20°C. The pressure ratio is 7. Maximum temperature in gas turbine cycle is 1000°C. In a supplementary firing, the gas temperature is raised to 1200°C. The steam generated in HRSG at 50 bar and 500°C. The condenser pressure is 0.1 bar. The temperature of gas leaving the HRSG is 200°C. The net power generated by CCPP is 345 MW. The isentropic efficiencies of compressor and gas turbine are 0.8 and 0.85 respectively. The calorific value of oil used is 45000 kJ/kg. Calculate : (i) air supplied to the compressor, (ii) combined cycle efficiency. Take, $C_{pa} = 1$ kJ/kg K, $\gamma = 1.4$ for air and $C_{pg} = 1.1$ kJ/kg K, $\gamma = 1.33$ for gas

[Ans : (i) 555 kg/s, (ii) 42%]

9.8 A combined power plant consists of a gas turbine unit and a steam turbine unit, the exhaust gases from the gas turbine is supplied to the steam generator. The pressure ratio for the gas turbine cycle is 10 and the inlet air temperature to the compressor is 18°C, and maximum cycle temperature for gas turbine is 1000°C. The temperature of gases leaving the steam generator is 160°C. The exhaust gases from gas turbine is heated further to 800°C before entering into HRSG. The condition of steam at steam turbine inlet is 50 bar and 500°C. The condenser pressure is 0.05 bar. The total power output of combined cycle plant is 200 MW. Isentropic efficiencies of air compressor, gas turbine and steam turbine are 82%, 84% and 85% respectively. Neglect pressure losses, pump work and effect of mass flow rate of fuel.
Calculate : (i) Mass flow rate of air and steam, (ii) Gas turbine cycle efficiency, (iii)

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Steam turbine cycle efficiency.

Take, $C_{pa} = C_{pg} = 1$ kJ/kg K, $\gamma = 1.4$ for both air and gas.

[**Ans** : (i) 501.7 kg/s, 97.3 kg/s, (ii) 28.4 % (iii) 33.4 %, (iv) 41.2 %]

Objective type questions

1. A gas turbine works on one of the following cycle...
   (a) Otto cycle (b) Diesel cycle (c) Rankine cycle (d) Brayton cycle
2. Overall efficiency of gas turbine plant is.....
   (a) more than diesel engine (b) more than petrol engine
   (c) less than diesel engine (d) none of these above
3. The maximum temperature in gas turbine may be about.....
   (a) 500 °C (b) 1000 °C (c) 1500 °C (d) 2000 °C
4. Constant pressure cycle gas turbine has .....
   (a) compression at constant pressure (b) heat addition at constant pressure
   (c) expansion at constant pressure
5. In a gas turbine cycle, the turbine output is 300 kJ/kg, the compressor work is 200 kJ/kg and heat supplied is 500 kJ/kg. The thermal efficiency of this cycle is.....
   (a) 20% (b) 40% (c) 60% (d) 80%
6. Which of the following parameter reduces thermal efficiency of gas turbine ?
   (a) decrease in compressor inlet temperature
   (b) increase in compressor efficiency
   (c) decrease in turbine inlet temperature
   (d) increase in turbine efficiency
7. Which of the following parameter increases air rate of gas turbine ?
   (a) decrease in compressor inlet temperature
   (b) increase in compressor efficiency
   (c) decrease in turbine inlet temperature
   (d) increase in turbine efficiency
8. Which of the following parameter does not increase with increase in pressure ratio up to optimum value ?
   (a) air rate (b) thermal efficiency
   (c) turbine work (d) compressor work
9. Water injection in the gas turbine decreases
   (a) power output (b) mass flow of air
   (c) thermal efficiency (d) work ratio
10. Normally, water is injected into working fluid of gas turbine at
    (a) before compression process (b) after compression process
    (c) after combustion process (d) after expansion process
11. Can type combustion chamber is widely used for industrial gas turbine but it is not

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suitable for aircraft engine due to ------

(a) large frontal area and weight
(b) difficult to control combustion
(c) difficulty in flame stabilization
(d) costly

12. The co-generation plant is produces .....

(a) only electric power
(b) only heat energy
(c) Both electric and heat energy
(d) none of these

13. Which of the following advantage/s is/are utilized in combined cycle power plant ?

(a) good part load performance
(b) quick starting and stopping
(c) low quality fuel can be used
(d) none of these

14. Following is an expression of efficiency of a combine cycle without secondary firing.

(a) $\eta = \eta_1 + \eta_2$
(b) $\eta = \eta_1 + \eta_2 - \eta_1\eta_2$
(c) $\eta = \eta_1 + \eta_2 + \eta_1\eta_2$
(d) $\eta = \eta_1 + \eta_2 - 2\eta_1\eta_2$

where η_1, η_2, η are efficiencies of topping, bottoming and combined cycle respectively.

15. Combined cycle power plants are gaining increasing acceptance as alternatives to conventional or nuclear steam power plants due to .....

(a) high thermal efficiency
(b) large power output capacity
(c) good part load performance
(d) none of these

16. In a combined cycle power plant, if load is reduced from 100 to 60%, the gas turbine output remains constant, the power output of plant is reduced by reducing power output of steam turbine due to following reason/s .....

(a) poor part load performance of gas turbine
(b) it is very difficult to control load on gas turbine
(c) poor part load performance of steam turbine
(d) all of these

17. Which of the following method is not improved the efficiency of combined cycle power plant.

(a) reducing inlet air temperature
(b) increasing exit gas temperature of gas turbine.
(c) operating the plant less than 60% load.
(d) increasing inlet gas turbine temperature.

Answer

1.(d) **2.** (c) **3.** (b) **4.** (b) **5.** (a) **6.** (c) **7.** (c) **8.** (a) **9.** (c) **10.** (b) **11.** (a)
12. (c) **13.** (b) **14.** (b) **15.** (a) **16.** (a) **17.** (c)

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GTU Study Materials ☑

Chapter 10
Nuclear Power Plant

10.1 Introduction
10.2 Nuclear fusion and fission
10.3 Chain reaction
10.4 Nuclear fuels
10.5 Components of nuclear reactor
10.6 Classification of reactors
10.7 Pressurized water reactor
10.8 Boiling water reactor
10.9 Gas cooled reactor
10.10 CANDU reactor
10.11 Liquid metal-cooled reactor
10.12 Fast breeder reactor
10.13 Nuclear waste and its disposal
10.14 Site selection for nuclear power plant
10.15 Nuclear power plants in India
10.16 Advantages and disadvantages of nuclear power plant

10.1 Introduction

As large amount of coal and petroleum are being used to produce energy, time may come when their reserves may not be able to meet the energy requirements. After each decade, the world demand for electricity is doubled due to booming increase in the population. Thus there is tendency to seek alternative sources of energy.

The unit cost per kWh of electricity generation in nuclear power plants is now comparable to or even lower than the unit cost in coal fired power plants in most parts of the world. Thermal power plants are coal based but the coal deposits are concentrated in the central and eastern parts of the India. In case of thermal power plants located away in other parts of country the coal has to be transported over the long distance which increases the cost of power generation. In addition, the problems associated with environmental pollution, mine safety, and so on are much less severe in nuclear power plants. Nuclear power utilization can help to save a considerable amount of fossil fuels which can be used in other areas of utility. Therefore the nuclear power will be the best alternative.

It has been found that one kilogram of uranium can produce as much energy as can be produced by burning 4000 tonnes of high grade coal or 1700 tonnes of oil. This shows that nuclear energy can be successfully employed for producing low cost energy in abundance as

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required by the expanding and industrialising population of future.

World's first atomic power plant of 5 MW capacity for industrial purposes was commissioned in former U.S.S.R on June 27, 1954 and this date is regarded as the birth of new source of power generation. The next nuclear Calder Hall power plant in Cumberland was commissioned in October 1956 to feed the electricity to the national grid of U.S.A. Many nuclear power stations are now operational throughout the world. In India also, the Nuclear Power Corporation has been forging ahead with installation of new plants all around the country.

In recent years, a strong public opinion has grown against the use of nuclear energy for power generation due to the problems related to nuclear safety, radioactive waste disposal and nuclear weapons proliferation.

10.2 Nuclear fusion and fission

Nuclear reactions of importance in energy production are fusion, fission, and radioactivity. In ***fusion***, *two or more light nuclei fuse to form a heavier nucleus. In fission a heavy nucleus is split into two or more lighter nuclei.* In both, there is a decrease in mass resulting in exothermic energy. This decrease in mass is called ***mass defect***. Most of the naturally occurring isotopes are stable. Those that are not stable called ***radioactive***. ***Radioactivity*** means that a radioactive isotope continuously undergoes spontaneous disintegration, usually with the emission of one or more smaller particles from the parent nucleus changing it into another, or daughter, nucleus. Radioactivity is always accompanied by a decrease in mass and is thus always exothermic.

- **Fusion :** **[Dec. '11]**

Energy is produced in the sun and stars by continuous fusion reactions. In this fusion process four nuclei of hydrogen fuse in a series of reactions and culminates in one nucleus of helium and two positron as given below :

$$4\,{}_1H^1 \rightarrow {}_2He^4 + 2\,{}_{+1}e^0$$

This results in a decrease in mass of about 0.0276 amu (atomic mass unit, 1 amu = 1.6604×10^{-24} grams) and thus release of 25.7 MeV (Million of electron volt – *An electron volt is the amount of energy required to move an electron charge a distance corresponding to a change in voltage from 0 to 1 volt*, $1J = 6.2316 \times 10^{18}$ eV). The heat produced in these reactions maintains temperature of the order of several million degrees in their cores and serves to trigger and sustain succeeding reactions.

On the earth, it is difficult to have fusion reactions for meaningful use due to various reasons. In order to cause fusion, it is necessary to accelerate the positively charged nuclei to high kinetic energies, to overcome electrical repulsive forces, by raising their temperature to hundreds of millions degree resulting in a plasma formation. Achieving such a high temperature is a difficult preposition. Further, plasma must be prevented from conducting the walls of the container and must be confined for a small period (about a second) at a minimum density. Fusion reactions are also called ***thermonuclear reactions*** as very high temperature are

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required to trigger and sustain them. Many problems have to be solved before artificially made fusion reactor becomes a reality.

- **Fission :** **[May '13, Dec. '10, Dec. '11]**

Fission can be caused by bombarding with high energy α-particles, protons, deuterons, X-rays as well as neutrons. However, neutrons are most suitable for fission, they are electrically neutral and thus require no high kinetic energy to overcome electrical repulsion from positively charged nuclei.

The nucleus of heavy elements like U-233, U-235, and Pu-239 has the ability to capture and absorb neutrons, whereupon it gets converted to a compound nucleus. Sometimes this compound nucleus is highly unstable undergoing spontaneous fragmentation into several approximately equal lighter nuclei plus two or three neutrons. This process is called ***nuclear fission*** and is accompanied by enormous amount of heat. Whereas isotopes U-238, Th-232 and Pu-240 are fissionable by neutrons of high energy (14 MeV) only.

As shown in Fig. 10.1, the immediate products of a fission reaction, such as Xe^{140}, and Sr^{94} are called ***fission fragments***, which along with other decay products (α, β, γ, etc.) are called ***fission products***.

$$_{92}U^{235} + {}_{0}n^{1} \longrightarrow {}_{54}Xe^{140} + {}_{38}Sr^{94} + 2\ {}_{0}n^{1} \qquad ...(10.1)$$

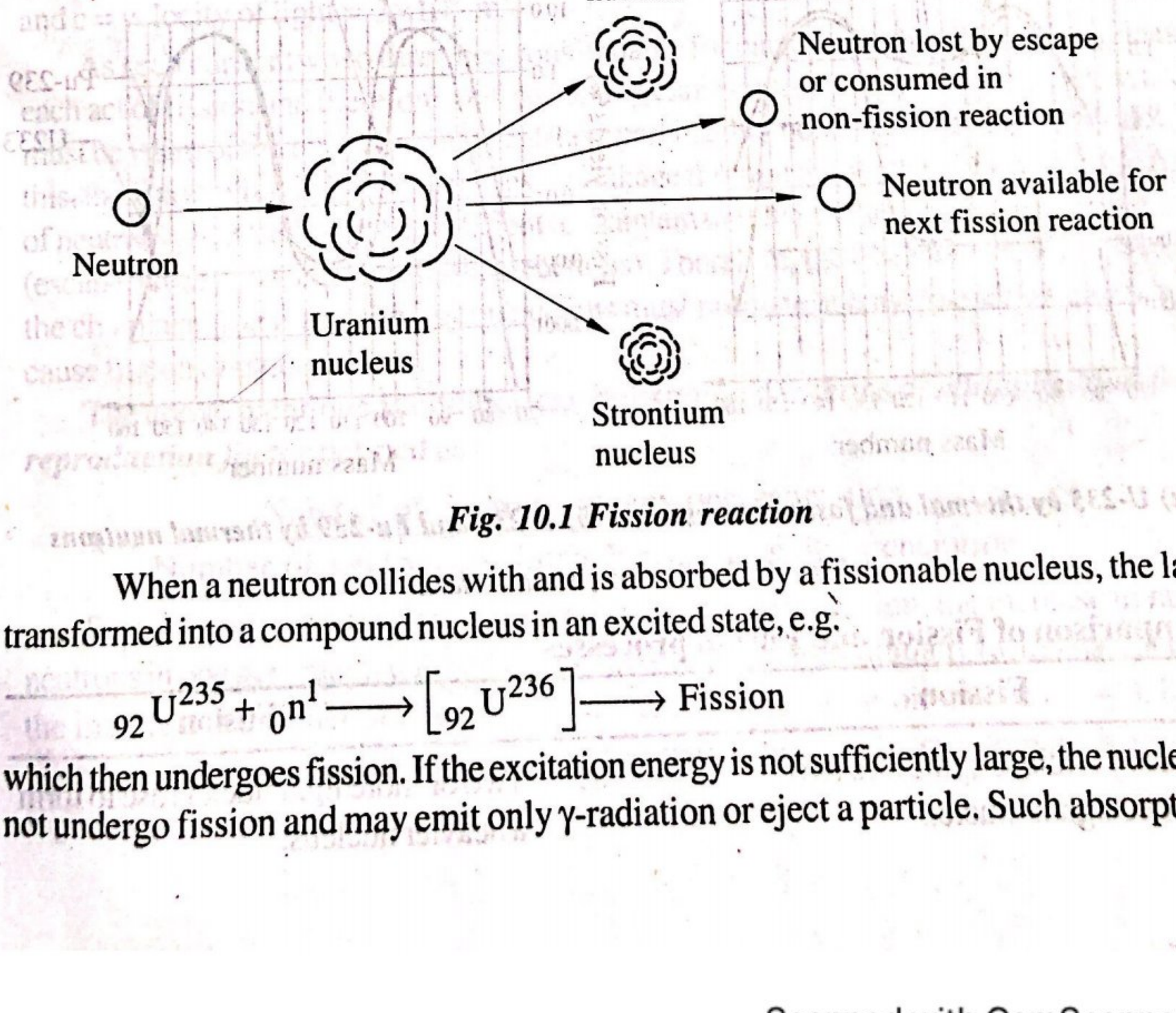


Fig. 10.1 Fission reaction

When a neutron collides with and is absorbed by a fissionable nucleus, the latter is transformed into a compound nucleus in an excited state, e.g.

$$_{92}U^{235} + {}_{0}n^{1} \longrightarrow \left[{}_{92}U^{236}\right] \longrightarrow \text{Fission}$$

which then undergoes fission. If the excitation energy is not sufficiently large, the nucleus may not undergo fission and may emit only γ-radiation or eject a particle. Such absorption of a

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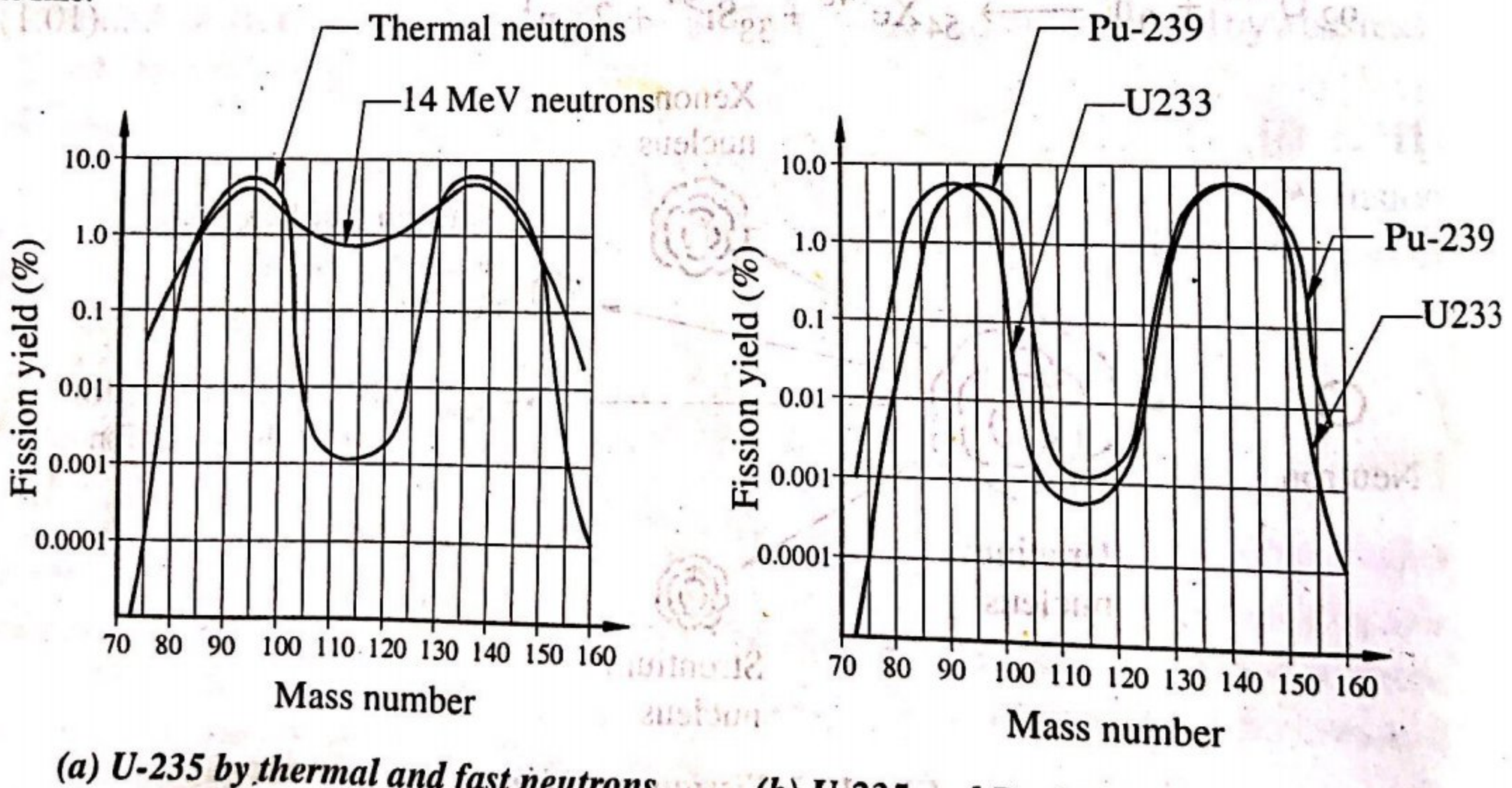
neutron in a non-fission reaction occurs about 16% of the time in all neutron absorptions by U-235.

The two fission fragments in Eq. (10.1) are not equal in size and are radioactive. The original nuclei of U-235 have neutron-proton ratio of 1.55. Their fission fragments have also similar n-p ratios, which for stable nuclei are, however, 1.2 to 1.4. The fission fragments, therefore, undergo several stages of β decay (converting neutrons into protons) until a stable product is formed in each case.

$$_{54}Xe^{140} \xrightarrow[16\,Sec]{\beta} {}_{55}Cs^{140} \xrightarrow[66\,Sec]{\beta} {}_{56}Ba^{140} \xrightarrow[12.8\,days]{\beta} {}_{57}La^{140} \xrightarrow[40\,hrs]{\beta} {}_{58}Ce^{140}\ (\text{stable})$$

$$\text{and}\quad {}_{38}Sr^{94} \xrightarrow[2\,min]{\beta} {}_{39}Y^{94} \xrightarrow[17\,min]{\beta} {}_{40}Zr^{94}\ (\text{stable}) \qquad \ldots(10.2)$$

This series is called a ***fission chain***. Since β-decay is usually accompanied by γ-radiation, suitable shielding against γ-rays as well as neutrons must be provided in a reactor. Fig.10.2 shows fission product data for U-235 by thermal and fast (14 MeV) neutrons and for U-233 and Pu-239 by thermal neutrons. The most probable fission products have mass numbers in the range 85 to 105 and 130 to 150, meaning that the products are not equal in size.



(a) U-235 by thermal and fast neutrons *(b) U-235 and Pu-239 by thermal neutrons*

Fig. 10.2 Fission product data

- **Comparison of Fission and Fusion processes**

Fission	Fusion
1. A heavy nucleus splits into two or more lighter nuclei.	1. Two or more light nuclei fuse to form a heavier nucleus.

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2. About one thousandth of the mass is converted into energy.	2. It is possible to have four thousandths of mass converted into energy.
3. Because of higher radioactive material, health hazards is high in case of accidents.	3. Because of lesser radioactive material health hazards is much less.
4. The process is possible at room temperatures.	4. The process is possible only at very high temperatures.
5. It is possible to construct self-sustained fission reactors and have positive energy release.	5. It is extremely difficult to construct controlled fusion reactor.
6. The links of the process are neutrons	6. The links of the process are protons.

10.3 Chain reaction

The essential condition for the practical utilization of nuclear energy is that a self-sustaining chain reaction should be maintained. Once the fission process is initiated in a few nuclei of fissile material, it should be able to continue throughout the remaining material without external influence.

The neutrons released have a very high velocity of the order of 1.5×10^7 metres per second. The energy liberated in the chain reaction is according to Einstein law

$$E = \Delta m c^2$$

where E = Energy produced in J, Δm = mass defect in kg.

and c = velocity of light = 3×10^8 m/s

As seen earlier two to three neutrons capable for causing further fission are released in each act of fission and therefore, self-sustaining chain-reaction is possible. However, account must be taken of the neutrons which can take part in other non-fission reactions. In addition to this, there is inevitable loss of neutrons by leakage through the system. Therefore, the number of neutrons produced must be sufficient to maintain the chain-reaction and to supply the losses (escape, absorption, and non-fission reactions). Therefore, the essential condition to maintain the chain reaction is that the fission nucleus must produce at least one secondary neutron to cause fission of other nucleus.

The above mentioned condition can be expressed in terms of ***multiplication factor*** or ***reproduction factor*** defined as

$$k = \frac{\text{Number of neutrons of any one generation}}{\text{Number of neutrons of immediately preceding generation}}$$

Since one neutron is required to maintain the chain reaction, the increase in number of neutrons in one generation will be $(k-1)$. If there are n neutrons initially (at the start of reaction); the increase in number of neutrons in the preceding generation will be $n(k-1)$. If t_1 is the average time between successive neutron generation, the rate of increase in neutron generation is given by

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$$\frac{dn}{dt} = \frac{n(k-1)}{t_1}$$

$$\therefore \frac{dn}{n} = \frac{k-1}{t_1} dt$$

$$\therefore \int \frac{dn}{n} = \int \left(\frac{k-1}{t_1}\right) dt$$

$\therefore \log_e n = \left(\frac{k-1}{t_1}\right) t + \log_e C$, where $\log_e C$ is a constant of integration.

$n = n_0$ at $t = 0$ $\quad \therefore C = n_0$

$$\therefore \log_e n = \left(\frac{k-1}{t_1}\right) t + \log_e n_0$$

$$\therefore \log_e \left(\frac{n}{n_0}\right) = \left(\frac{k-1}{t_1}\right) t$$

$$\therefore n = n_0 (e)^{(k-1)t/t_1} \quad \ldots(10.3)$$

It is obvious from the above equation if $k > 1$, the number of neutrons increases exponentially with time.

Suppose in a particular case $k = 1.005$ and $t_1 = 0.001$ sec. (average life time for neutron), the number of neutrons after one second will increase by

$$\frac{n}{n_0} = e^{\left(\frac{0.005 \times 1}{0.001}\right)} = e^5 = 150 \text{ times}$$

If $k < 1$, the chain reaction cannot be maintained as neutron production decreases in an exponential manner.

The neutrons are lost during the fission leakage, non-fission capture by U^{238} and U^{235} (known as resonance capture) and non-fission capture by moderator, coolant, structural material, and fission products.

If say $n_0 = 100$ and $k = 2$, then the fission causes to release 200 fission neutrons in the next succeeding fission. If 100 neutrons out of 200 are lost by the ways mentioned above and 100 are available for next fission then the chain reaction is possible. If the loss is more than 100, the reaction dies out.

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When $k < 1$, the system is known as ***subcritical*** (stopping the reactor) and when $k >$ is the system is known as ***super-critical*** (atom-bomb) and when $k = 1$, the system is known as ***critical*** and this is the desirable requirement for power reactors.

The smaller the surface-volume ratio of the reactor core; i.e. the large its size, the lower is the percentage of leakage of neutrons. The core size is increased to the point where a chain reaction is possible. This size is called the ***critical size***. The mass of fuel in such a core is called the ***critical mass***. Therefore, *the size of a uranium reactor is determined by nuclear considerations of criticality and not by the power output.*

10.4 Nuclear fuels

The nuclear fuels which are generally used in reactor are ${}_{92}U^{235}$, ${}_{94}Pu^{239}$ and ${}_{92}U^{233}$.

Natural uranium ore consists of three isotopes of uranium as ${}_{92}U^{238}$, ${}_{92}U^{235}$, and ${}_{92}U^{234}$. In natural uranium the availability of U^{238} is largest upto the extent of 99.28%, U^{235} is only 0.715% which is most unstable and fissionable and the remainder 0.006% is U^{234}. In ${}_{92}U^{238}$ number 92 represent 92 protons, number 238 represent its mass nunnber and the difference $(238 - 92) = 146$ represent the number of neutrons in it.

The other two fuels ${}_{94}Pu^{239}$ and ${}_{92}U^{233}$ are formed in the nuclear reactors during fission process from ${}_{92}U^{238}$ and ${}_{90}Th^{232}$ respectively due to the absorption of neutron without fission.

Fissionable materials *are those which are capable of sustaining a fission chain reaction*. U-235 is the only fissionable isotope found in nature.

Fertile materials *are non-fissionable materials that can be converted into fissionable materials*. U-238 and Th-232 can be converted to fissionable materials. *They are the fertile materials.*

The chain reaction cannot be maintained in some reactors with natural uranium having only 0.7% fissionable U^{235}. Therefore, it is necessary to increase the percentage of U^{235} in the fuel if used in reactor for maintaining the chain reaction. *The process used to increase the percentage of* U^{235} *is known as* ***enrichment***. The enrichment required may lie from 1.5% to 90% according to its use in a particular reactor. The enrichment of fuel also reduces size of the reactor.

The practical methods for enrichment are based on the slight difference in the atomic masses of U^{238} and U^{235}. The cost of uranium enrichment is high.

The different methods which are used for the enrichment are listed below :

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(1) Gaseous diffusion method
(2) Thermal diffusion method
(3) Centrifugal method
(4) Electro-magnetic method.

It is estimated that almost all the resources of uranium are situated in U.S.A. (33%), South Africa (20%), Australia (20%) and Canada (20%). The largest reserve of most economical and low cost uranium lies in Australia.

In nuclear reactions, the reactant nuclei do not show up in the products, instead we may find either isotopes of the reactants or other nuclei. However, the number of nucleons in the reactants and products remain the same in a chemical reaction. For example, consider the reactants M and N and the products O and P in a nuclear reaction as follows.

$$_{z_1}M^{A_1} + _{z_2}N^{A_2} \rightarrow _{z_3}O^{A_3} + _{z_4}P^{A_4}$$

To balance this reaction, the following relationship must be satisfied

$$z_1 + z_2 = z_3 + z_4, \; A_1 + A_2 = A_3 + A_4$$

10.5 Components of nuclear reactor [May'13]

The nuclear reactor may be regarded as a substitute for the boiler fire box of coal fired steam power plant or combustion chamber of gas turbine plant. The heat produced

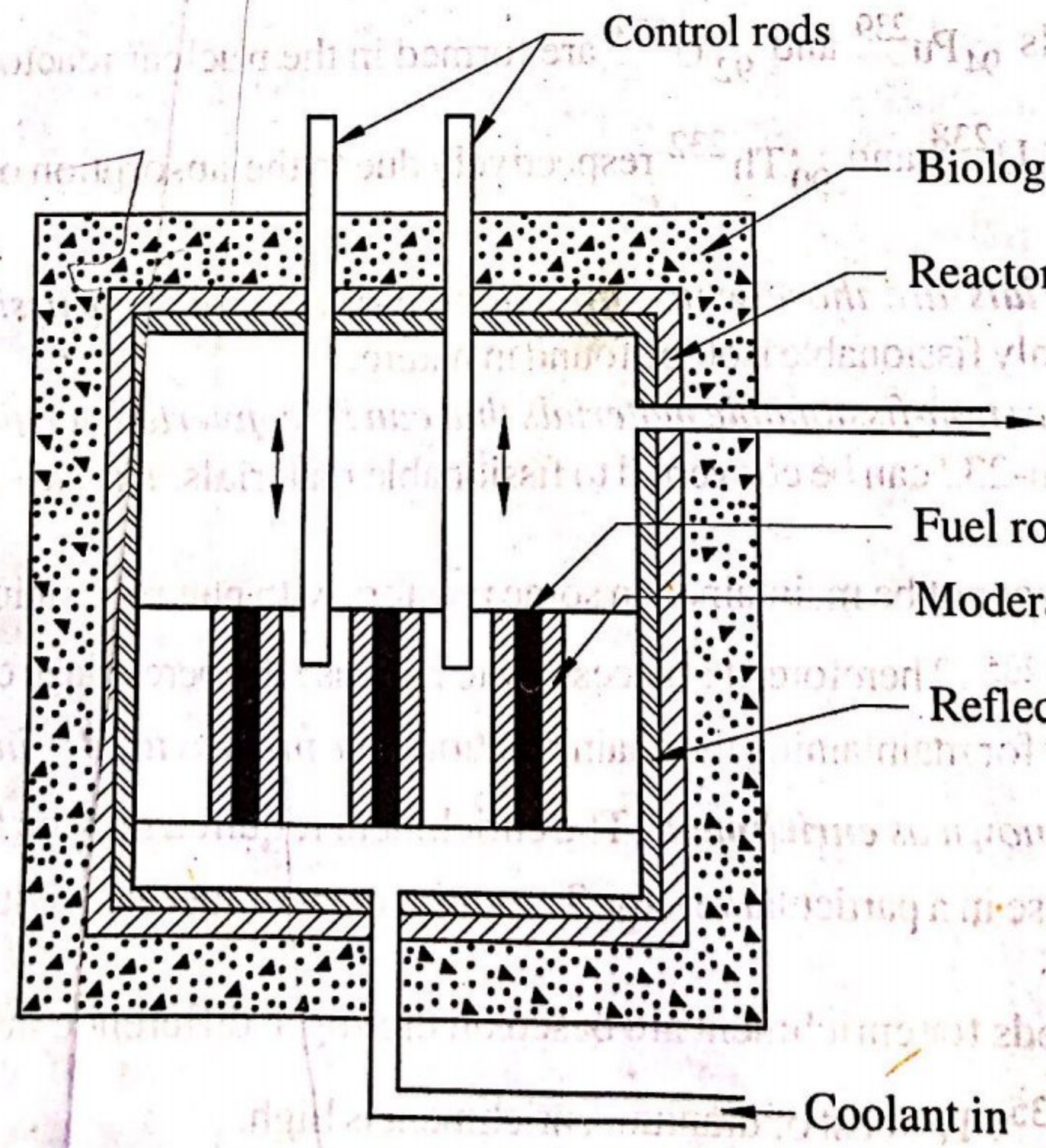


Fig. 10.3 Components of Nuclear reactor

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in the nuclear reactor is by fission process whereas in steam and gas power plants, the heat is produced by combustion of fuel. The other cycle of operation and components required is the same either as steam plant if steam is generated by using the heat released due to fission or a gas turbine plant, if gas is heated by using the heat released due to fission. The steam or gas may be the working fluid in nuclear power plant.

The general arrangement of nuclear reactor with principle components is shown in Fig. 10.3.

(1) Fuel

Nuclear reactors use fissionable materials like ${}_{92}U^{235}$, ${}_{94}Pu^{239}$ and ${}_{92}U^{233}$. Natural uranium found in earth's crest has 0.714% of ${}_{92}U^{235}$. ${}_{92}U^{235}$ is unstable and capable of sustainable chain reactions.

The fuel is shaped in various shapes like rods, plates, pallets, tins, etc. and located in the reactor in such a manner that the heat production within the reactor is uniform. The fuel elements are designed taking into account the heat transfer, corrosion and structural strength.

In ***homogeneous reactors***, the fuel and moderator are mixed to form a uniform mixture i.e. uranium and carbon and then it is used in the form of rods or plates in the reactor core. In ***heterogeneous reactors***, the fuel is used in the form of rods or plates and moderator surrounds the fuel elements. This arrangement is commonly used in the most of the reactors.

The fuel rods are clad with stainless steel, zirconium or by aluminium to prevent oxidation of uranium.

(2) Moderator **[June '11, Dec.'11]**

It is a material used to slow down the neutrons from high kinetic energy (1 MeV or 13200 km/s) to slow neutron (0.25 eV or 2200 m/s) in a fraction of a second. The function of the moderator is to increase the probability of reaction. The fission chain reaction in the nuclear reactor is maintained due to slow neutrons when the ordinary uranium is used as fuel.

The moderators are lighter than fuel. Various materials used for moderators are ordinary water (H_2O), deutorium or heavy water (D_2O), graphite and beryllium. The graphite, heavy water or beryllium can be used as moderator with natural uranium. The ordinary water (H_2O) is used as moderator only with the enriched uranium. As a moderator D_2O is the best material having moderating ratio of 12000 as compared to 72 for H_2O and 170 for carbon.

The following are the desirable properties of a good moderator :

(1) It must be as light as possible, the slowing down action is more effective in elastic collision with light element.

(2) It must not absorb neutrons during slowing down them.

(3) It must be able to work under high pressure and high temperature with good corrosion resistance.

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(4) It must have high chemical stability as well as it should not be decomposed due to the nuclear radiation.

(3) Control rods **[June '11, Dec. '11]**

The control system controls the rate of energy generated. It starts, increases, decreases and stops the reaction. These rods may be shaped like the fuel rods themselves and are interspread throughout the reactor core. Instead of containing fuel, they contain neutron absorber such as boron, cadmium or indium. They have high neutron-absorption cross-sections.

The control is necessary to prevent the melting of fuel rods, disintegration of coolant and destruction of reactor as the amount of energy released is enormous.

The control rods can be moved in and out of the reactor core assembly either automatically or manually. The magnitude of insertion of control rod represent the proportionate absorption of neutrons.

(4) Coolant

The main purpose of the coolant in the reactor is to transfer the heat produced in the reactor and to keep the fuel assembly at a safe temperature to avoid their melting and destruction. The same heat carried by the coolant is used in the heat exchanger for further utilization in the power generation either generating steam or using hot gas.

The coolant used should have high specific heat to reduce its mass flow rate. It should not absorb neutrons and it must be non-corrosive, non-toxic, non-oxidising with high chemical stability.

The water, heavy water, gas (He, CO_2), sodium in liquid form and organic liquids are used as coolant.

(5) Reflector

It is important to conserve the neutrons as much as possible for reducing consumption of fissile material and to keep the size of the reactor small. This is possible by surrounding the reactor core with a material which reflects escaping neutrons back into the core. This material is called reflector. The required properties of a good reflector are low absorption and high reflection for neutrons, high resistance to oxidation and irradiation as well as high radiation stability.

The materials used for moderators are also used for reflectors e.g. H_2O, D_2O and carbon. It is necessary to provide some method of cooling the reflector as it gets heated due to collision of neutrons with its atoms.

(6) Shielding

Shielding the radioactive zones in the reactor from possible radiation hazard is essential to protect the human life from the harmful effects. The common radiations from the reactor are α-rays, β-rays, γ-rays and fast neutrons. To prevent the effects of these radiations on the human life, it is necessary to absorb them before emitting to the atmosphere.

Shielding consists of inner lining of 50 to 60 cm thick steel plate on the reactor core called ***thermal shield*** with a few meters of thick concrete wall surrounding the inner shield called

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biological shield. The lining of steel plate absorbs these energies and becomes heated but prevents the adjacent wall of reactor vessel from becoming heated. The thermal shield is cooled by the circulation of water.

(7) Reactor vessel

The reactor vessel encloses the reactor core, reflector and shield. It also provides coolant inlet and outlet passages. The reactor vessel has to withstand the pressure at 200 bar or above. The reactor core (fuel and moderator assembly) is generally placed at the bottom of the vessel.

10.6 Classification of reactors

The nuclear reactors can be classified on the following basis.

(1) On the basis of Neutron energy :

(a) Fast reactors : In these reactors, the fission is affected by fast neutrons without any use of moderators.

(b) Thermal reactors : In these reactors the fast moving neutrons are slowed down by passing them through the moderator. These slow moving neutrons are then captured by the fuel material to bring about the fission. The moderator is most essential component in these reactors.

(c) Intermediate reactors : In these reactors, the velocity of neutrons is kept between fast reactors and thermal reactors.

(2) On the basis of fuel used :

(a) Natural uranium fuel reactors : In this reactor, the natural uranium is used as fuel.

(b) Enriched uranium fuel reactors : In this reactor, the uranium used contains 5 to 10% of U^{235}.

(3) On the basis of coolant used :

(a) Water cooled reactors
(b) Heavy water cooled reactors
(c) Gas cooled reactors
(d) Liquid metal (sodium or bismuth) cooled reactors
(e) Organic liquid cooled reactors

(4) On the basis of moderator used :

(a) Water moderated
(b) Heavy water moderated
(c) Graphite moderated
(d) Beryllium moderated

(5) On the basis of reactor core used :

(a) Homogeneous reactors
(b) Heterogeneous reactors

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10.7 Pressurized water reactor [Dec. '10, June '11]

A Pressurized Water Reactor (PWR) is a thermal reactor where slightly or highly enriched, heterogeneous, clad uranium is used as fuel and ordinary water under high pressure is used as a moderator, coolant, and reflector.

The arrangement of pressurised water reactor is shown in Fig. 10.4. The pressurising tank included in the circuit maintains the constant pressure in the circuit throughout the load range. The pressurizer is simply a pressure vessel with a heater at the bottom and a water spray at the top. The water in the pressurizer is boiled and converted into steam by electric heating coil before starting the reactor. This results in the increase in pressure of the primary circuit. If the primary circuit pressure becomes too high, cold water is sprayed into the steam in the pressurizer. This condenses the steam and reduces the primary circuit pressure.

The water in the primary circuit gets heated by absorbing the fission energy in the reactor core and same energy is given in the heat exchanger to generate the steam in secondary circuit. In order to prevent the boiling of water in the core, it is kept under pressure of about 130 to150 bar. It helps in absorbing the heat by water in the liquid state in the reactor. The steam produced in secondary circuit is used in a conventional way in the steam power plant cycle as shown in Fig. 10.4. The water coolant from heat exchanger is recirculated to the reactor with the help of coolant pump.

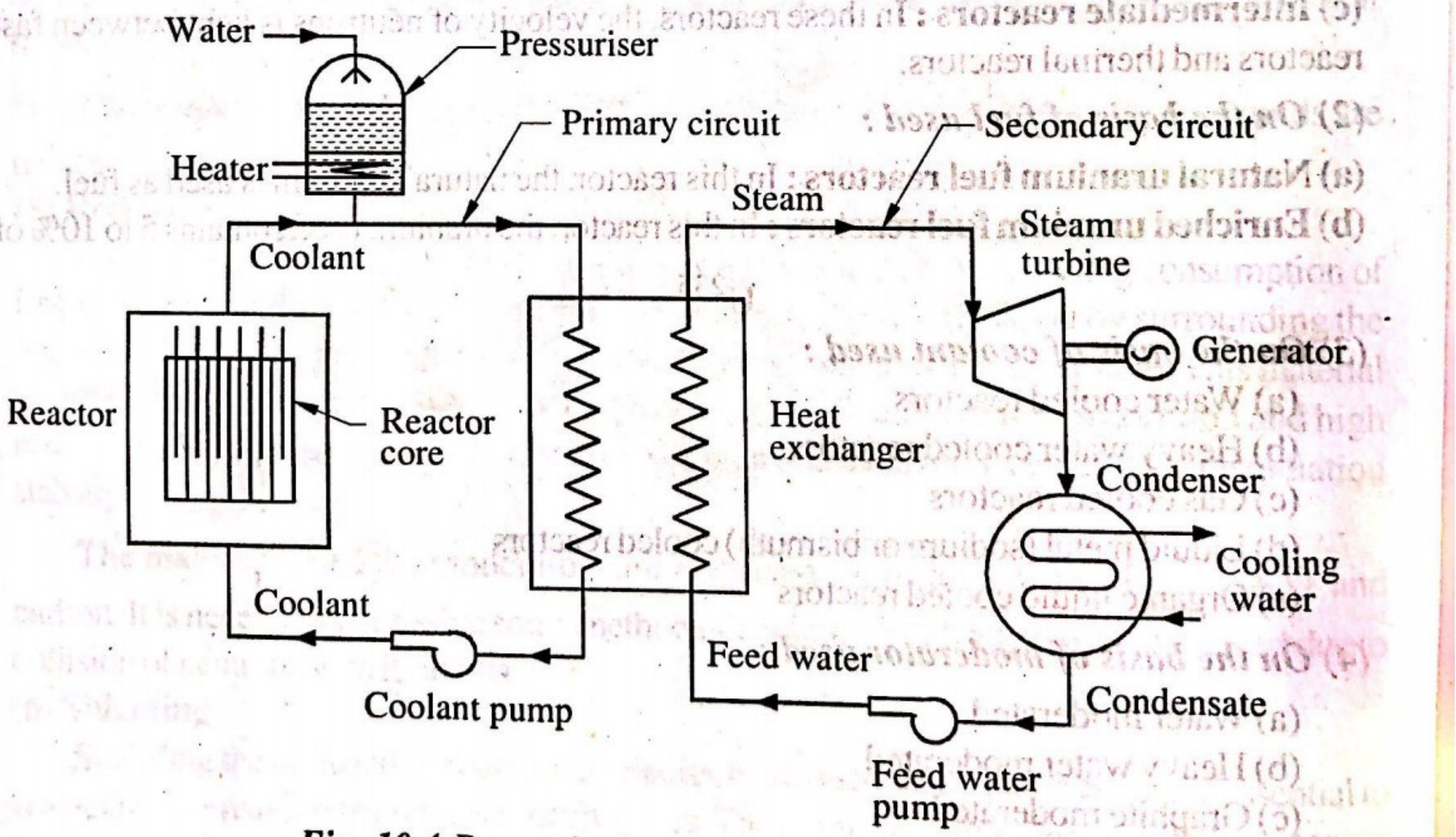


Fig. 10.4 Pressurized water reactor power plant

In this reactors, the water flowing through the reactor becomes radioactive, therefore this primary circuit must be heavily shielded to protect the operators.

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- **Advantages :**
  (1) It uses ordinary water as a coolant, moderator and reflector since water is available abundantly and cheaply, there is great saving in cost.
  (2) The reactor is very compact and has high power density.
  (3) As steam is not contaminated by radiation, inspection and maintenance of the turbine, condenser and feed pump is possible.
  (4) Small number of control rods are required.
  (5) Greater energy is extracted per unit weight of fuel loaded.
- **Disadvantages :**
  (1) It's capital cost is high due to heavy and strong pressure vessel is required in the primary circuit.
  (2) The thermodynamic efficiency of this plant is as low as 20% due to low pressure in the secondary circuit.
  (3) It requires enriched uranium fuel, cost of enriched uranium is high.
  (4) It is necessary to shut down the reactor for fuel charging.
  (5) The corrosion problems are more severe as the corrosion is accelerated in the presence of high pressure, high temperature water.

10.8 Boiling water reactor [June '11, Dec. '11]

In Boiling Water Reactor (BWR), enriched uranium is used as fuel and water is used as coolant, moderator and reflector like PWR except the steam is generated in the reactor itself instead of separate steam generator.

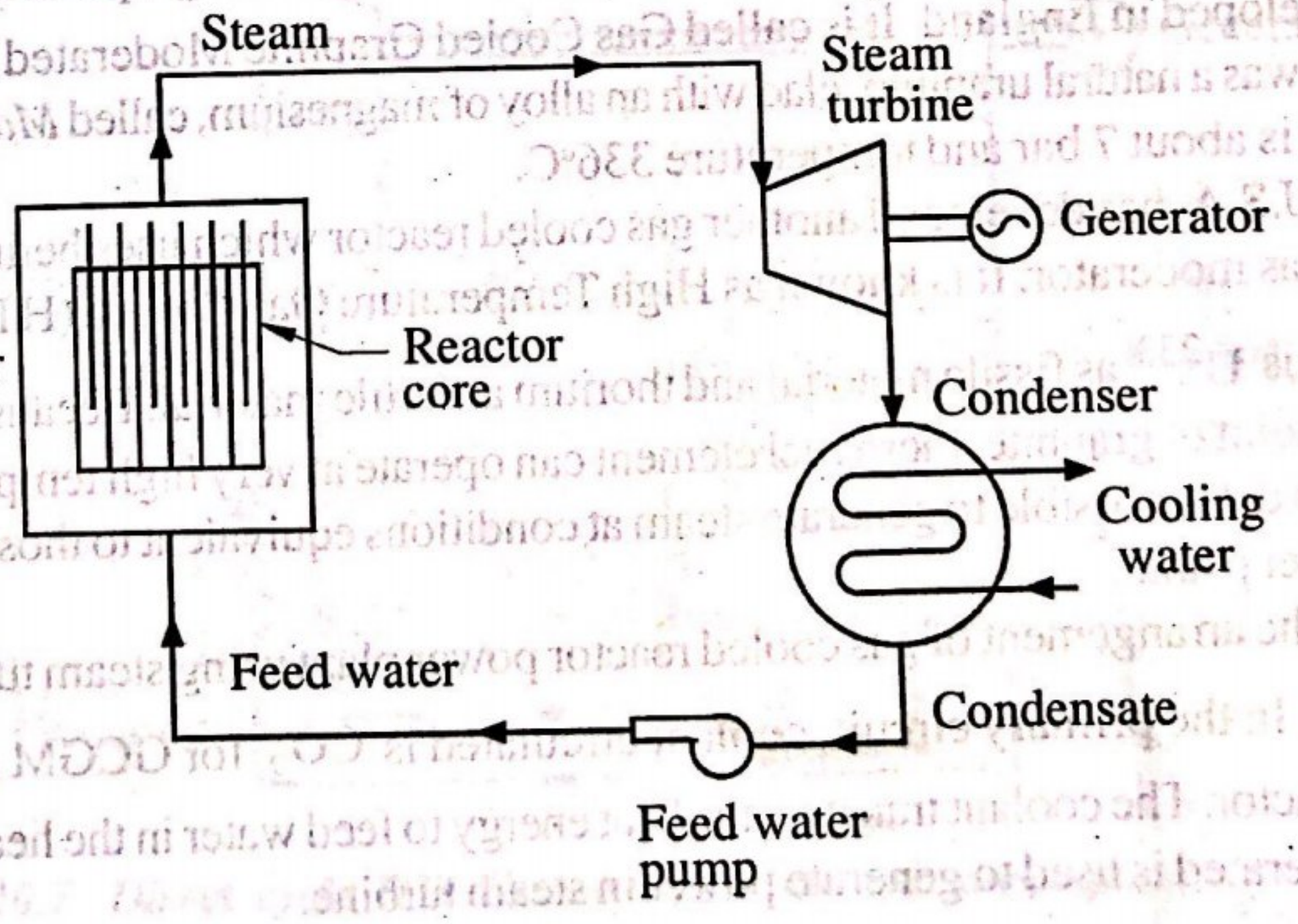


Fig. 10.5 Boiling water reactor

The arrangement of the boiling water reactor power plant is shown in Fig. 10.5. The feed water circulated in the reactor is converted into saturated steam by transfer of heat released by

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fission in the reactor core. This steam is supplied to steam turbine working on conventional power plant cycle. The mechanical power produced by the turbine is converted into electric energy by the generator.

- **Advantages :**
  (1) The pressure inside the reactor vessel is less than PWR as water is allowed to boil inside the reactor. Therefore, the reactor vessel can be much lighter than PWR and reduces the cost of pressure vessel considerably.
  (2) It eliminates the use of heat exchanger, pressuriser, circulating pump and piping. Therefore, the cost is further reduced.
  (3) The thermal efficiency of this reactor plant (30%) is considerably higher than PWR plant.
  (4) The metal temperature remains low for given output conditions.
- **Disadvantages :**
  (1) The steam leaving the reactor is slightly radioactive. Therefore, light shielding of turbine and piping is necessary.
  (2) It cannot meet the sudden changes in load on the plant.
  (3) The power density of this reactor is nearly 50% of PWR, therefore, the size of the vessel will be considerably large compared with PWR.
  (4) It requires enriched uranium as a fuel.

10.9 Gas cooled reactor [Jan. '13]

The gas cooled reactors with CO_2 gas as coolant and graphite as moderator were first developed in England. It is called Gas Cooled Graphite Moderated (GCGM) reactor. The fuel was a natural uranium, clad with an alloy of magnesium, called *Magnox*. The coolant pressure is about 7 bar and temperature 336°C.

U.S.A. has developed another gas cooled reactor which uses helium as coolant and graphite as moderator. It is known as High Temperature Gas Cooled (HTGC) reactor. The fuel used is U^{233} as fissile material and thorium as fertile material. Because of the very high melting point of graphite, these fuel element can operate at very high temperature (700°C to 800°C) and it is possible to generate steam at conditions equivalent to those in modern coal-fired power plant.

The arrangement of gas cooled reactor power plant using steam turbine is shown in *Fig. 10.6*. In the primary circuit, coolant circulated is CO_2 for GCGM reactor or He for HTGC reactor. The coolant transfers the heat energy to feed water in the heat exchanger. The steam generated is used to generate power in steam turbine.

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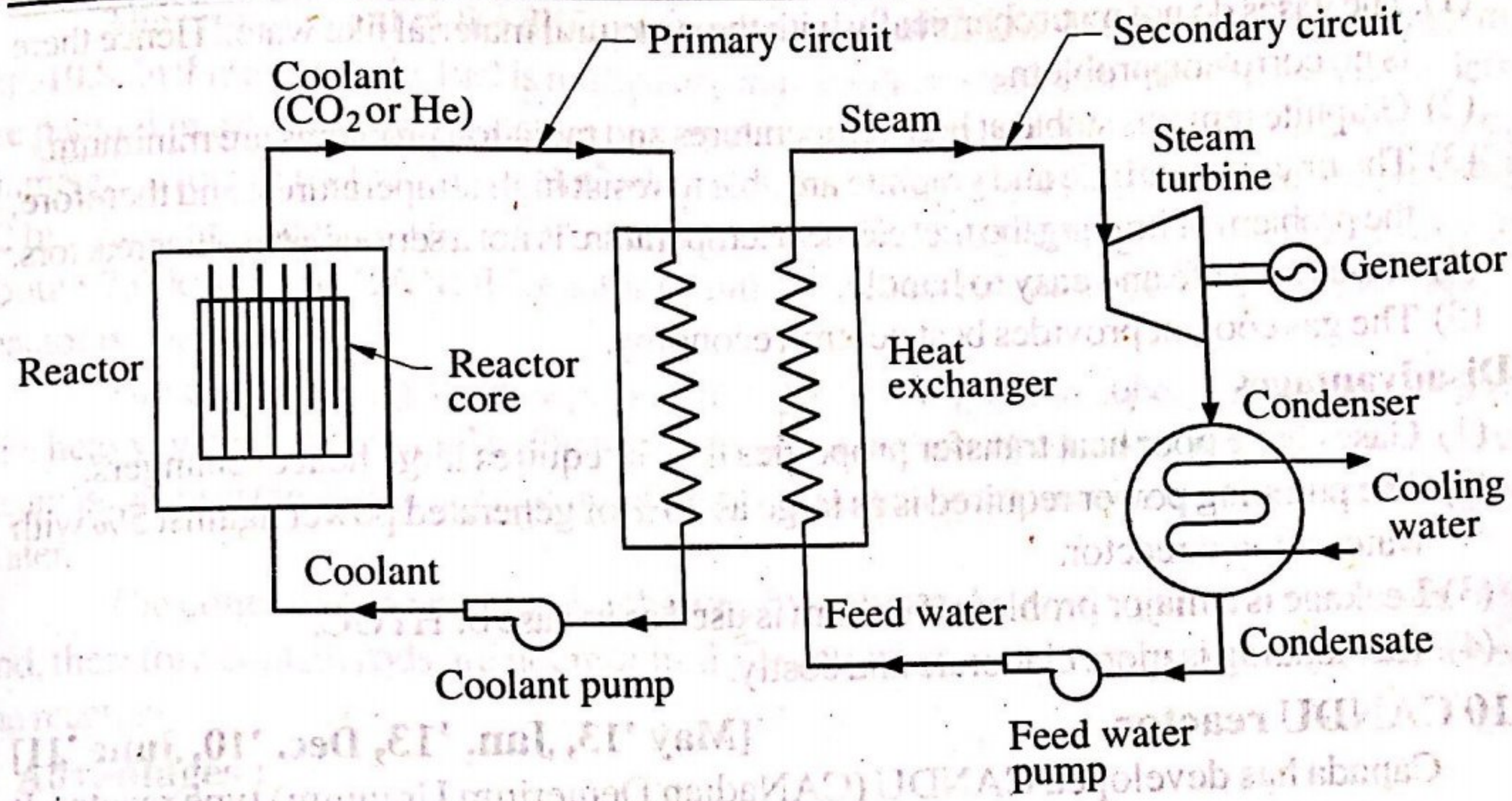


Fig. 10.6 Gas cooled reactor power plant

Fig. 10.7 shows the arrangement of HTGC reactor direct cycle Gas turbine plant. This type of plant uses gas turbine for power production and coolant as helium gas. It consists a regenerator, precooler and compressor. The fuel used may be U-233 or Th-232.

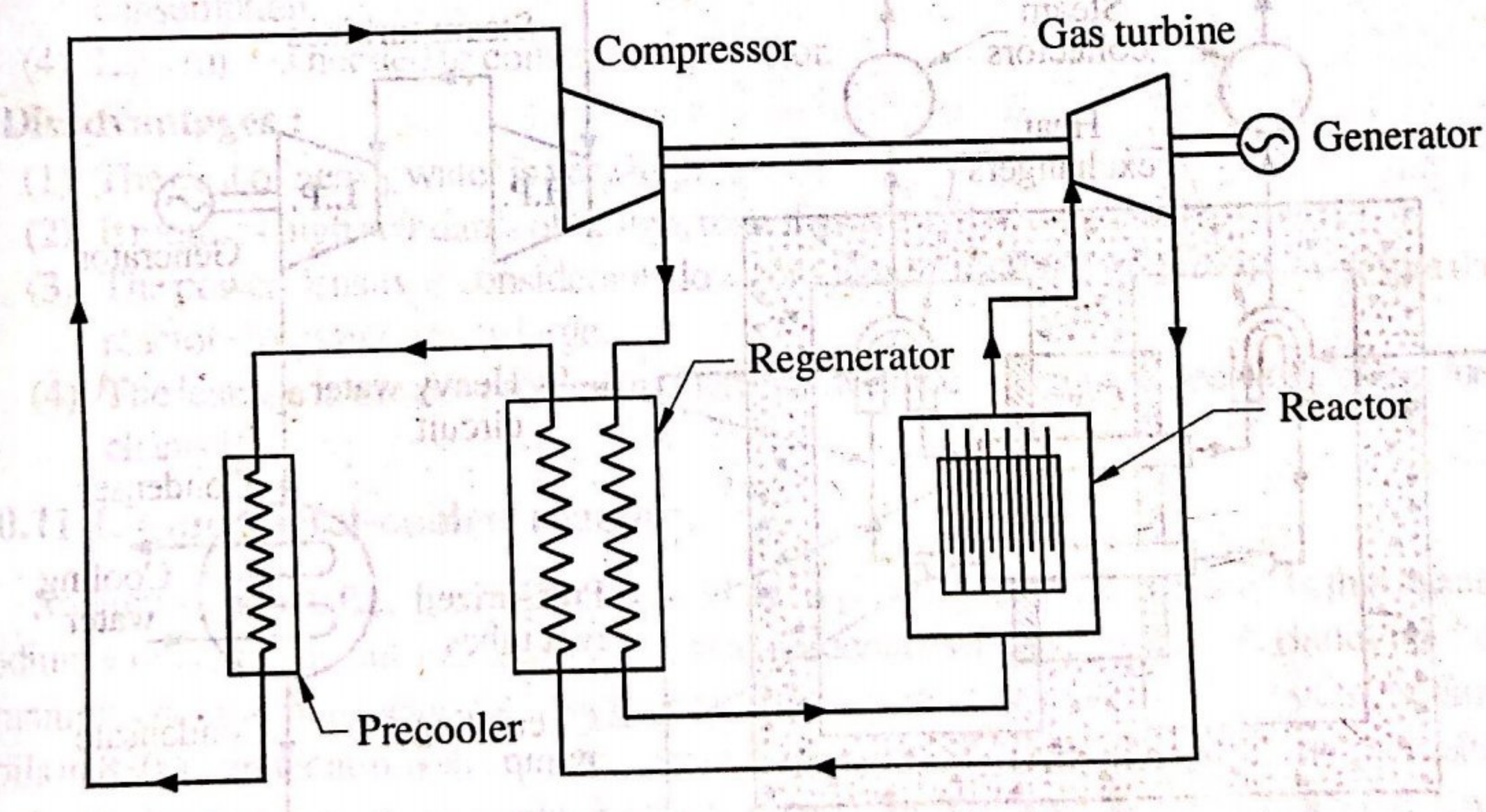


Fig. 10.7 Direct cycle HTGC reactor gas turbine power plant

- **Advantages**

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(1) The gases do not react chemically with the structural material like water. Hence there is no corrosion problem.
(2) Graphite remains stable at high temperatures and radiation problems are minimum.
(3) The uranium carbide and graphite are able to resist high temperatures, and therefore, the problem of limiting the fuel element temperature is not a serious as in other reactors.
(4) Gases are safe and easy to handle.
(5) The gas coolant provides best neutron economy.

- **Disadvantages**

(1) Gases have poor heat transfer properties thus it requires large heat exchangers.
(2) The pumping power required is as large as 20% of generated power against 5% with water-cooled reactor.
(3) Leakage is a major problem if helium is used as in case of HTGC.
(4) Fuel loading is more elaborate and costly.

10.10 CANDU reactor [May '13, Jan. '13, Dec. '10, June '11]

Canada has developed CANDU (CANadian Deuterium Uranium) type reactor. It uses Pressurized Heavy Water (PHW) (deuterium oxide, D_2O) as moderator and coolant while the fuel used is natural uranium. These reactors are more economical to those nations which do not produce enriched uranium as the enrichment of uranium is very costly.

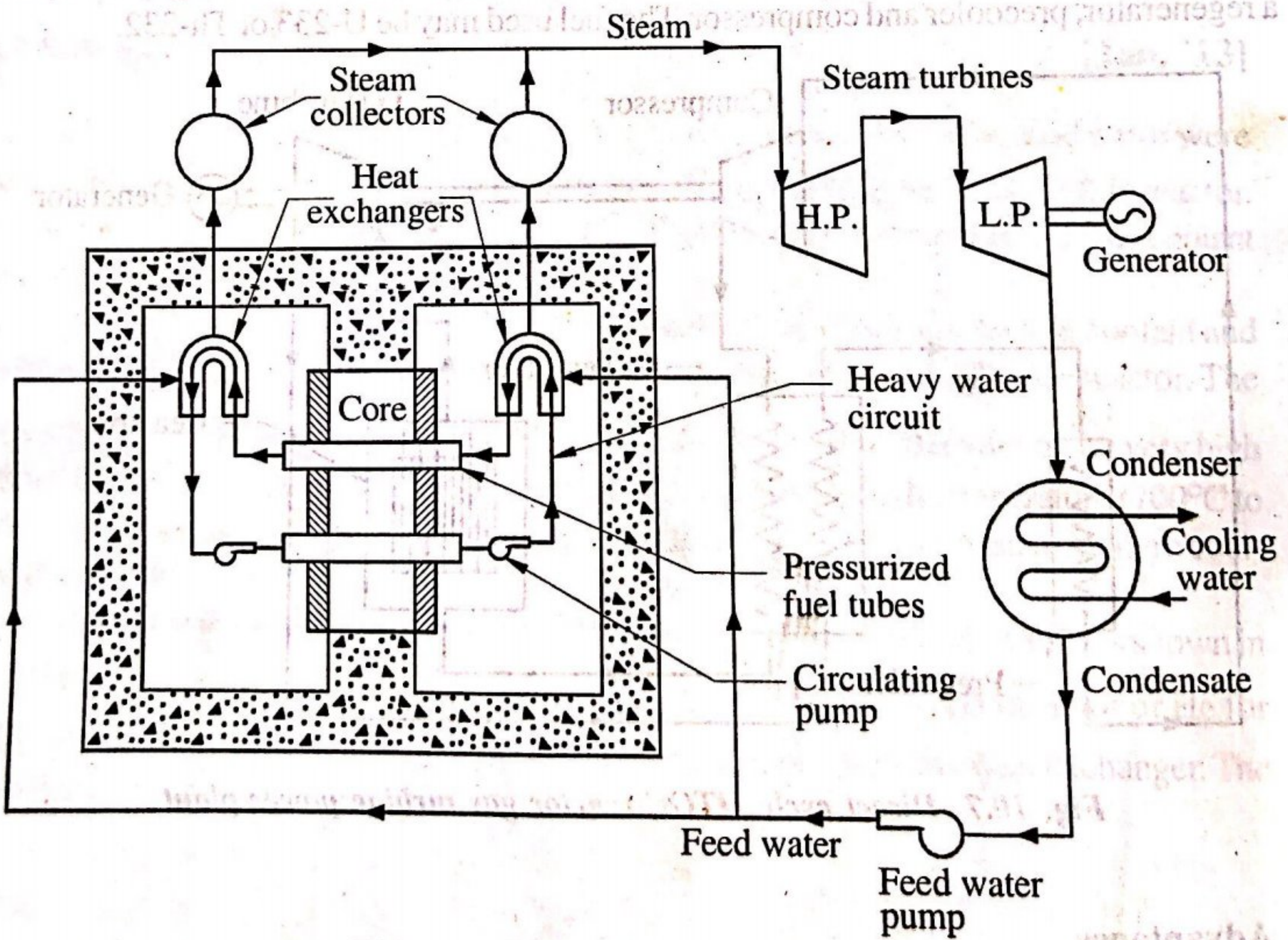


Fig. 10.8 CANDU reactor power plant

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The arrangement of the different components of CANDU type reactor is shown in Fig. 10.8. In this reactor, the fuel is normal uranium oxide as small cylinder pellets. The pellets are packed in a corrosion resistance zirconium alloy tube, nearly 0.5 cm long and 1.3 cm diameter, to form a fuel rod. The relatively short rods are combined in bundles of 37 rods, and 12 bundles are placed end to end in each pressure tube. The total mass of fuel in the core is about 97,000 kg. The CANDU reactor is unusual in that refueling is conducted while the reactor is operating.

The coolant heavy water is passed through the fuel pressure tubes and heat exchanger. The heavy water is circulated in the primary circuit in the same way as with a PWR and the steam is raised in the secondary circuit transferring the heat in the heat exchanger to the ordinary water.

The control of the reactor is achieved by varying the moderator level in the reactor and, therefore control rods are not required. Cadmium rods are used to start and shut down the reactor.

- **Advantages :**
  (1) Fuel enrichment is not required.
  (2) The reactor vessel may be built to withstand low pressure compared with PWR and BWR. Only the fuel tubes are designed to withstand high pressure, therefore, the cost of the vessel is less.
  (3) Heavy water is used as moderator, which has higher multiplication factor and low fuel consumption.
  (4) Less time is needed to construct the reactor.
- **Disadvantages :**
  (1) The cost of heavy water is very high.
  (2) It requires high standards of design, manufacture and maintenance.
  (3) The power density is considerably low compared with PWR and BWR, therefore the reactor size is extremely large.
  (4) The leakage is a major problem as there are two mechanically sealed closures per fuel channel.

10.11 Liquid metal-cooled reactor

Sodium-Graphite Reactor (SGR) is one of the typical liquid metal reactors. In this reactor sodium works as a coolant and graphite works as moderator while the fuel used is little enriched uranium. The common metals which can be used as coolant are sodium and potassium. Sodium boils at 880°C under atmospheric pressure and freezes at 95°C. Hence sodium is first melted by electric heating system and be pressurised to about 7 bar. The liquid sodium is then circulated by circulation pump.

The arrangement of the components of a metal cooled reactor is shown in Fig. 10.9. The reactor will have two coolant circuits or loops. In the primary circuit the heat is absorbed by liquid sodium in the reactor. The sodium becomes radioactive while it passes through the core

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and reacts chemically with water. Therefore, the heat absorbed by sodium is transferred to secondary coolant sodium potassium (NaK) in the primary heat exchanger which in turn transfers the heat in the secondary heat exchanger called steam generator. The steam generated from this steam generator will be superheated.

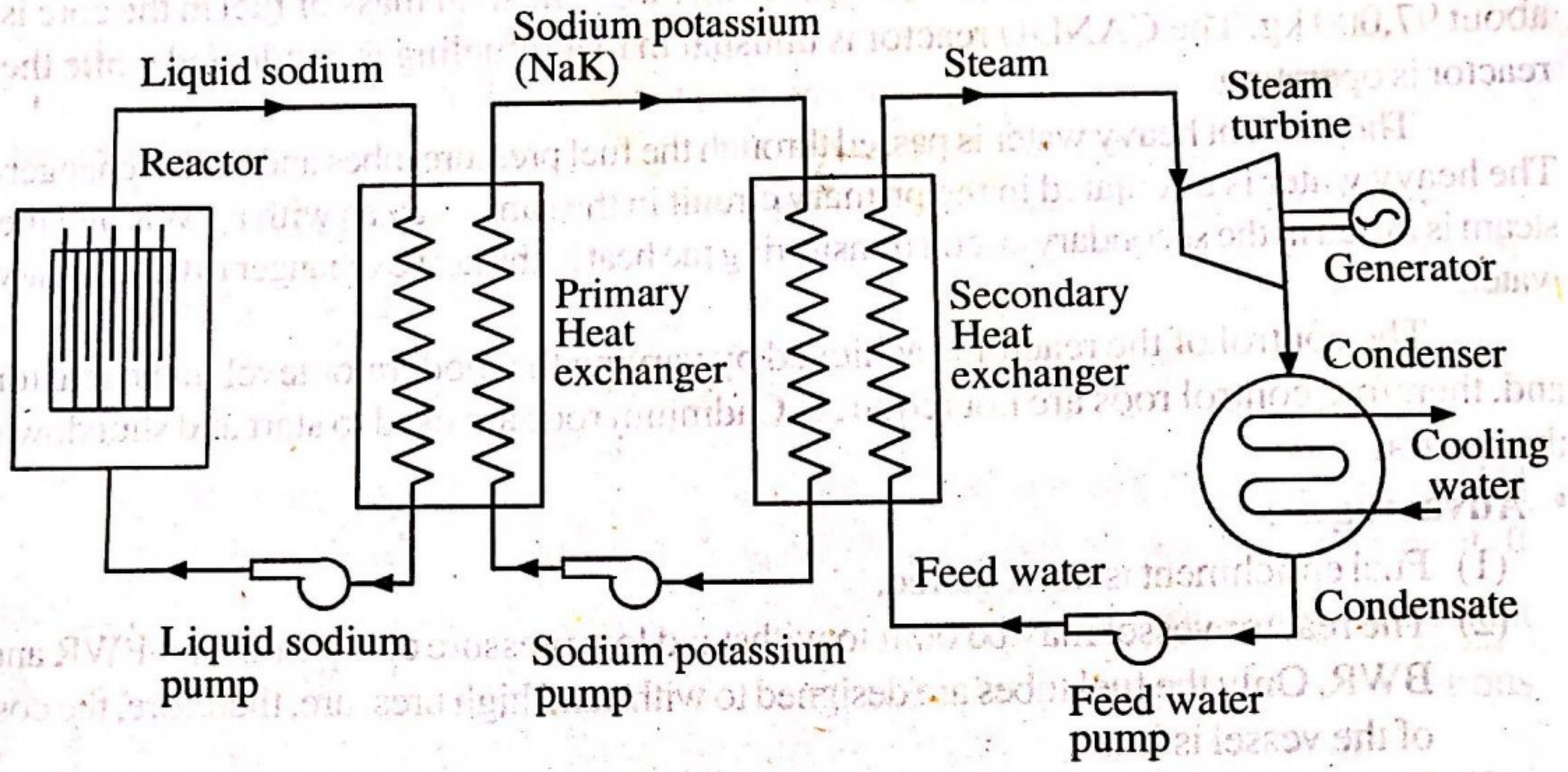


Fig. 10.9 Liquid metal reactor

The reactor vessel, primary circuit and primary heat exchanger have to be shielded from radiations. The liquid metal is to be handled under the cover of an inert gas such as helium, to prevent contact with air while charging or draining the primary or secondary circuit.

- **Advantages :**
  (1) High temperatures can be achieved in the cycle and that means high thermal efficiency.
  (2) The sodium as a coolant need not be pressurised.
  (3) The reactor size is comparatively small.
  (4) The low cost graphite moderator can be used as it can retain its mechanical strength and purity at high temperatures.
  (5) Superheating of steam is possible.
- **Disadvantages :**
  (1) Sodium reacts violently with water and actively with air.
  (2) Intermediate heat exchanger is necessary to separate radioactive sodium from water.
  (3) It is necessary to shield the primary and secondary cooling systems with concrete blocks as sodium becomes highly radioactive.
  (4) The leak of sodium is very dangerous compared with other coolants as it comes out of reactor in highly radioactive state.

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10.12 Fast breeder reactor [Dec. '13]

The process of converting more fertile material into fissile material in a reactor is known as ***breeding***. Its importance lies in consequence that all available fertile material can be converted into fissile material.

In fast breeder reactor, the core containing U^{235} is surrounded by a blanket of fertile material U^{238}. In this reactor no moderator is used. The fast moving neutrons liberated due to fission of U^{235} are absorbed by U^{238} which gets converted into Pu^{239} (plutonium), a fissile material. This man made fuel Pu^{239} can be used for further fission. Similarly, thorium (Th^{232}) can be converted into U^{233} which is also a secondary fissile material. Therefore, these type of reactors are important since they not only produce heat but also produce more secondary fissile fuels than fuel consumed in the reactor. Since India has massive reserves of thorium and limited resources of uranium, development of these fast breeder reactors are important for our country.

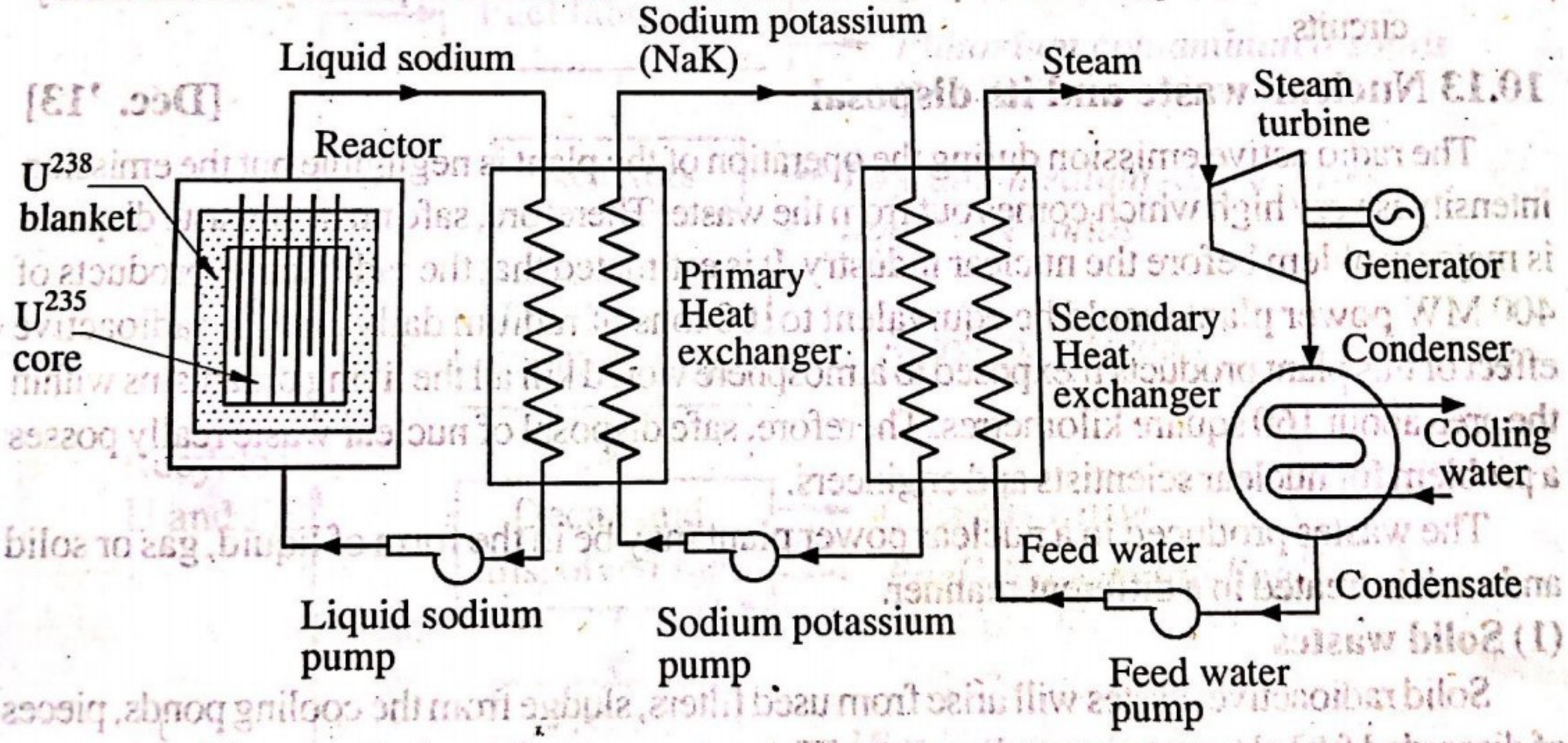


Fig. 10.10 Liquid metal fast breeder reactor

The arrangement of liquid metal fast breeder reactor is shown in Fig. 10.10. The enriched uranium U^{235} or Pu^{239} is kept without a moderator in the reactor core surrounded by the thick blanket of depleted U^{238}. One additional neutron available from fission of U^{235} is used to convert U^{238} or Th^{232} into Pu^{239} or U^{233} as secondary fuels. This reactor also uses two liquid metal coolants in which sodium is used as primary coolant and sodium potassium as secondary coolant. Liquid sodium is circulated through reactor to carry the heat

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produced. The heat absorbed by sodium is transferred to secondary coolant sodium potassium (NaK) in the primary heat exchanger which in turn transfers the heat in the secondary heat exchanger called steam generator. The major difficulty with fast breeder reactor is to remove the large quantities of heat from the core as the power density is very high which is 40 times greater than CANDU type reactor, 13 times greater than BWR and 200 times greater than gas cooled reactor. Therefore, special coolants and special arrangements are necessary to carry out large quantity of heat.

- **Advantages :**
  (1) It gives high power density than any other reactor, therefore, small core is sufficient.
  (2) Moderator is not required.
  (3) Secondary fissible materials by breeding are obtained.
  (4) Absorption of neutron is slow.
- **Disadvantages :**
  (1) It requires highly enriched (15%) uranium fuel.
  (2) Safety must be provided against melt down.
  (3) Neutron flux is high at the centre of the core.
  (4) Thick shielding is necessary against radioactive radiations in primary and secondary circuits.

10.13 Nuclear waste and its disposal [Dec. '13]

The radio active emission during the operation of the plant is negligible but the emission intensity is very high which comes out from the waste. Therefore, safe nuclear waste disposal is major problem before the nuclear industry. It is estimated that the radioactive products of 400 MW power plant would be equivalent to100 tons of radium daily and the radioactive effect of this plant products if exposed to atmosphere would kill all the living organisms within the area about 160 square kilometres. Therefore, safe disposal of nuclear waste really posses a problem for nuclear scientists and engineers.

The wastes produced in a nuclear power plant may be in the form of liquid, gas or solid and each is treated in a different manner.

(1) Solid wastes

Solid radioactive wastes will arise from used filters, sludge from the cooling ponds, pieces of discarded fuel element cans, splitters etc. These along with discarded items of the plant. e.g. control rods.

There are many ways for disposing of the solid fission products. The storing in shielded storage vaults consists in fixing the solid waste in borosilicate glass and then storing this glass in leak tight capsules. These capsules or vaults can then be stored in deep salt mines or in deep wells drilled in the stable ocean floor. The combustible solid waste is burnt in incinerators and the flue gases formed are filtered and disposed off through stacks. Active solid wastes are stored in water for about 100 or more days to allow radioactivity to decay. Then these are disposed off to deep salt mines or on ocean floor or in deep wells drilled in stable geological strata.

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(2) Liquid wastes

Radioactive liquid effluents will arise from the laundry, personal decontamination, etc. together with activity accumulating from the corrosion of the irradiated fuel elements in the storage ponds.

The disposal of liquid waste is done in two ways :

(a) Dilution : The liquid wastes are diluted with large quantities of water and then released into the ground. This method suffers from the drawback that there is a chance of contamination of under ground water if the dilution factor is not adequate.

(b) Concentration to small volumes and storage : When the dilution of radioactive liquid wastes is not desirable due to amount or nature of isotopes, the liquid wastes are concentrated to small volumes and stored in underground tanks. The tanks should be of assured long term strength and leakage of liquid from the tanks should not take place, otherwise leakage of contents from the tanks may lead to significant underground water contamination.

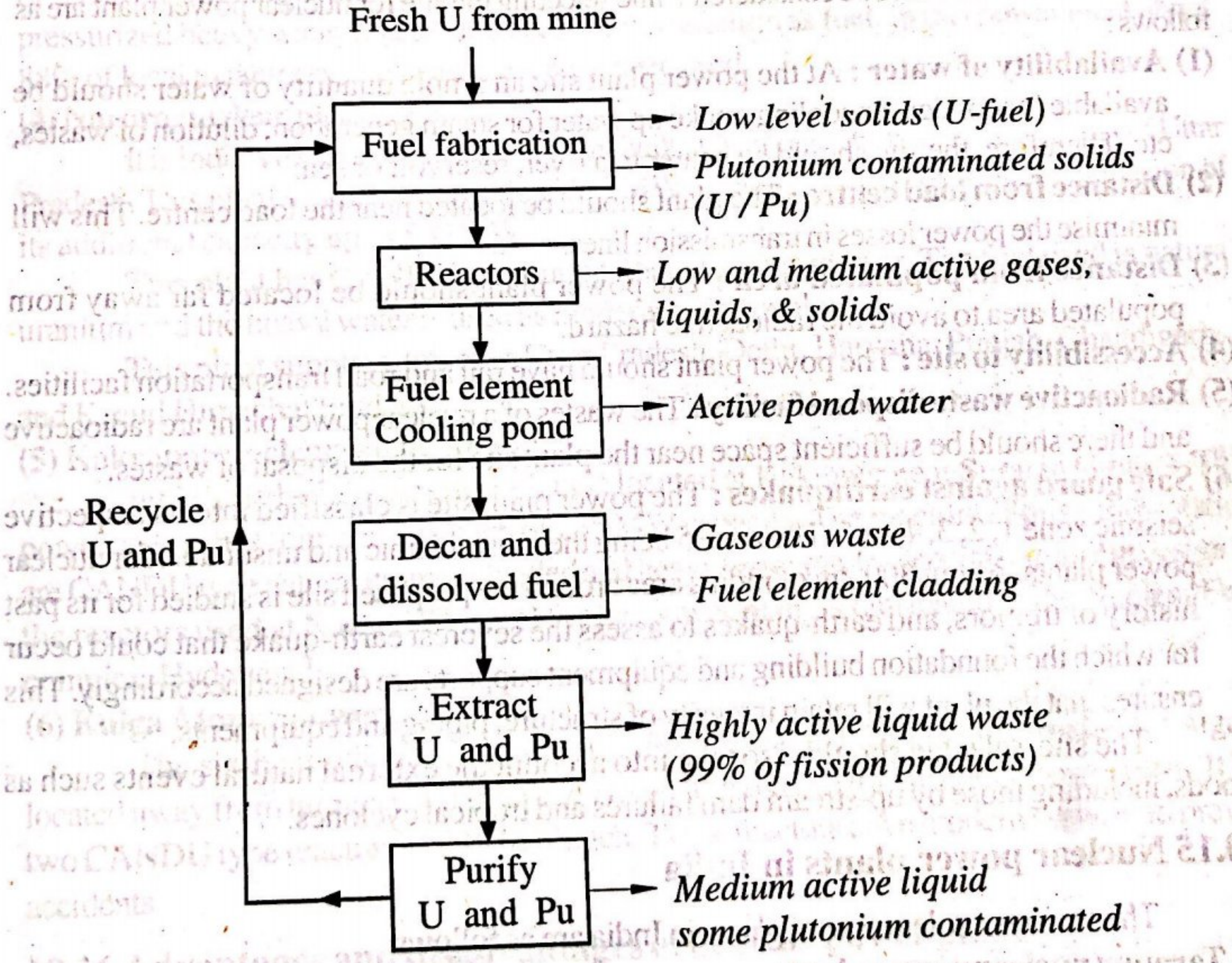


Fig. 10.11 Nuclear fuel cycle

(3) Gaseous wastes :

During normal operation the following discharges of radio-active gaseous coolant will occur :

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- Leakage from the pressure circuit at blower oil seals, from charge and discharge machines.
- Krypton and iodine gases coming out from spent nuclear fuel, etc.

Gaseous wastes are generally diluted with air passed through filters and then released to atmosphere through large chimneys. Krypton is removed by cryogenic treatment of dissolved off gas stream and packed in gas cylinders under pressure. Iodine can be removed from off-gases by caustic scrubbing or HNO_3 scrubbing.

Generally radioactive gases are collected and stored in the tank buried in the ground and disposed off to the atmosphere when activity level sufficiently comes down.

The outline of nuclear fuel cycle is shown in Fig. 10.11 and different types of wastes coming out at different stages are also labelled on the Fig. 10.11.

10.14 Site selection for nuclear power plant

The various factors to be considered while selecting the site for nuclear power plant are as follows :

(1) **Availability of water :** At the power plant site an ample quantity of water should be available for condenser cooling, make up water for steam generation, dilution of wastes, etc. Therefore, the site should be nearer to a river, reservoir or sea.

(2) **Distance from load centre :** The plant should be located near the load centre. This will minimise the power losses in transmission lines.

(3) **Distance from populated area :** The power plant should be located far away from populated area to avoid the radioactive hazard.

(4) **Accessibility to site :** The power plant should have rail and road transportation facilities.

(5) **Radioactive waste disposal facility :** The wastes of a nuclear power plant are radioactive and there should be sufficient space near the plant site for the disposal of wastes.

(6) **Safe guard against earthquakes :** The power plant site is classified into its respective seismic zone 1, 2, 3, 4, or 5. The zone 5 being the most seismic and unsuitable for nuclear power plants. About 300 km of radius area around the proposed site is studied for its past history of tremors, and earth-quakes to assess the severest earth-quake that could occur for which the foundation building and equipment supports are designed accordingly. This ensures that the plant will retain integrity of structure, piping and equipments.

The site selected should also take into account the external natural events such as floods, including those by up-stream dam failures and tropical cyclones.

10.15 Nuclear power plants in India

The various nuclear power plants in India are as follows

(1) Tarapur nuclear power plant :

It is India's first nuclear power plant. It has been built at Tarapur, 100 km north of Mumbai, in collaboration with U.S.A. It has two boiling water reactors (BWR) each of 200 MW capacity and uses enriched uranium as its fuel and ordinary water as coolant and moderator.

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It first unit was commissioned in 1961 and the second unit in 1966. It supplies power to Maharashtra and Gujarat.

The cost of the total plant was Rs. 65 crores (on the basis of price line in 1964), out of that Rs. 45.25 crores was financed by U.S.A. Government on repay basis within 40 years.

(2) Rana Pratap Sagar nuclear power plant :

It is the second nuclear power plant of India and was installed in 1971. It has been built at 67 km south-west of Kota in Rajasthan in collaboration with Canada. It has two reactors of CANDU type each of 200 MW capacity. It uses natural uranium as fuel in the form of oxide and heavy water as moderator and coolant.

(3) Kalpakkam nuclear power plant :

It is the third nuclear power plant in India and built at about 65 km away from Chennai, Tamil Nadu state. It is the first nuclear power plant which has been wholly designed and constructed by Indian scientists and engineers using indigenous materials. It has two units each of 235 MW capacity. Its first unit was installed in 1983 and the other unit in 1988. It has pressurized heavy water reactor and uses natural uranium as fuel. In this power plant about 88% of local machinery and equipment have been used.

(4) Narora nuclear power plant

It is India's fourth nuclear power plant built at Narora in Bullandshahar district of Uttar Pradesh. This plant has two units each of 235 MW capacity with a provision for extension of its additional capacity up to 500 MW.

This plant has CANDU-pressurized heavy water reactor. The fuel used is natural uranium and the heavy water is used as moderator and coolant.

This plant supplies power to Uttar Pradesh, Delhi, Haryana, Punjab, Chandigarh, J and K, and Himachal Pradesh.

(5) Kakrapar nuclear power plant :

India's fifth nuclear power plant is located at Kakrapar near Surat in Gujarat. This power station has four reactors each of 235 MW capacity. The reactors of this power station are CANDU type natural uranium fuelled and heavy water moderated. It is similar in design as the reactors used at Narora. The fuel for the power plant is fabricated at the Nuclear Fuel complex, Hyderabad.

(6) Kaiga Atomic power plant :

The sixth atomic power plant of India is located at Kaiga in Karnataka. Kaiga is located away from human habitation and is a well suited site for an atomic power plant. It has two CANDU type reactors of 235 MW each. These reactors have modern systems to prevent accidents.

10.16 Advantages and disadvantages of nuclear power plant.

The various advantages and disadvantages of a nuclear power plant over thermal power plants are as follows :

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- **Advantages :**
  (1) There are no fuel transportation, handling and storage charges since nuclear fuel requirements is much less compared to coal. Further the nuclear power plants conserve the fossil fuels for other energy need.
  (2) The nuclear power plant is more economical compared with thermal in areas which are remote from coalfields.
  (3) The nuclear power plant occupies less space in comparison to thermal plants, therefore, the civil construction cost is also less.
  (4) The capital cost is low for large sized power plants and the running cost is competitive to the thermal plants.
  (5) Less number of persons are required for its operation. Therefore, the cost of operation is less.
  (6) There is increased reliability of operation.
  (7) No ash handling problem.
  (8) These plants are non-polluting unlike coal or oil fired power plants.
- **Disadvantages :**
  (1) These plants are not suitable for variable load requirements.
  (2) Radioactive wastes if not disposed carefully may have bad effect on the health of workers and other population.
  (3) The preservation of radioactive waste for a long time creates lot of difficulties and requires huge capital.
  (4) It requires trained personnel to operate nuclear power plant.
  (5) Maintenance cost of the plant is high.
  (6) The danger of radioactivity always persists in the nuclear power plants.

Exercise

10.1 What do you understand by nuclear fusion and fission ?
10.2 What do you understand by radioactivity ?
10.3 What are fission fragment and fission product ? Explain fission reaction with an example.
10.4 Give comparison of fission and fusion processes.
10.5 Explain fission chain reaction.
10.6 What do you mean by critical size and critical mass ?
10.7 What is the difference between fissionable and fertile materials ?
10.8 What do you mean by uranium enrichment ?
10.9 Define Reproduction factor (k).
10.10 Give the functions and materials used for following components of nuclear reactor : (i) fuel, (ii) moderator, (iii) reflector, (iv) control rod, and (v) biological shield.
10.11 Give the functions and materials used for following components of nuclear reactor : (i) coolant, (ii) thermal shield, (iii) reactor vessel.

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10.12 State the main components of a nuclear reactor.
10.13 What is a fast reactor ? What is a thermal reactor ?
10.14 What are homogeneous and heterogeneous reactors ?
10.15 Explain with neat sketch Pressurized Water Reactor (PWR). Explain function of pressurizer in PWR.
10.16 Explain the characteristic features of a PWR.
10.17 Write short note on Nuclear Reactor.
10.18 Explain with neat sketch Boiling Water Reactor (BWR).
10.19 Compare PWR and BWR.
10.20 What are Gas Cooled Reactors (GCR) ? State the types of GCR and the difference between them. State the advantages and disadvantages of GCR.
10.21 Explain direct cycle High Temperature Gas cooled (HTGC) reactor gas turbine power plant with neat sketch.
10.22 Explain with neat sketch construction and working of CANDU type reactor.
10.23 State the fuel, moderator and coolant used in the CANDU type reactor.
10.24 Explain with neat sketch construction and working of liquid metal-cooled reactor.
10.25 What is breeding ? Explain the characteristic features of fast breeder reactor.
10.26 Explain fast breeder reactor with neat sketch.
10.27 Explain Nuclear waste and its disposal.
10.28 State any four nuclear power plants in India including the power plant situated in Gujarat.
10.29 State the advantages of fast breeder reactor.
10.30 Discuss any four nuclear power plants situated in India.
10.31 Explain site selection for nuclear power plant.
10.32 State the advantages and disadvantages of nuclear power plant.

Objective Type Questions

1. In the nuclear fission reactions _____ isotope of uranium is used.
   (a) U^{238} (b) U^{234} (c) U^{235} (d) all of the above
2. The nuclear energy is measured as
   (a) MW (b) MeV (c) Curie (d) Fermi
3. Two or more light nuclei fuse to form a heavier nucleus, this process is known as
   (a) fusion (b) fission (c) breeding (d) all of the above
4. Which of the following are fertile materials
   (a) U^{233} and Pu^{239} (b) U^{238} and Th^{232}
   (c) U^{238} and Th^{239} (d) U^{238} and Pu^{239}
5. When a nuclear reactor is operating at constant power, the multiplication factor (k) is
   (a) greater than unity (b) less than unity
   (c) equal to unity (d) none of the above.

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6. Critical mass of fuel is the amount required to make the multiplication factor
   (a) equal to unity (b) more than unity
   (c) less than unity (d) none of the above
7. Which of the following is not fissionable material.
   (a) U^{235} (b) Pu^{239} (c) U^{233} (d) U^{238}
8. The function of the moderator in a nuclear reactor is to
   (a) absorb neutrons (b) stop chain reaction
   (c) reduce temperature (d) reduce the speed of neutrons
9. Following is the most commonly used moderator
   (a) Sodium (b) Graphite (c) Deuterium (d) NaK
10. In a nuclear reactor the function of a reflector is to
   (a) absorb neutrons
   (b) reduce the speed of the neutrons
   (c) reflect the escaping neutrons back into the core
   (d) stop cain reaction
11. Thermal shielding is provided to
   (a) protect the operating personnel from exposure to radiation
   (b) reduce the speed of the neutrons.
   (c) reduce temperature (d) all of the above.
12. Reflectors of a nuclear reactor are made up of
   (a) graphite (b) steel (c) boron (d) beryllium
13. In a pressurised water reactor (PWR)
   (a) the pressurised water is work as moderator, coolant, and reflector
   (b) the pressurised water is work as moderator and coolant only
   (c) the pressurised water is work as coolant only.
   (d) the pressurised water boils in the core of the reactor.
14. A CANDU reactor uses
   (a) enriched uranium as fuel and ordinary water as coolant and moderator.
   (b) natural uranium as fuel and ordinary water as coolant and moderator.
   (c) natural uranium as fuel and heavy water as coolant and moderator.
   (d) enriched uranium as fuel, heavy water as coolant and graphite as moderator.
15. Following nuclear reactor does not require moderator
   (a) pressuried water reactor (b) fast breeder reactor
   (c) boiling water reactor (d) CANDU type reactor
16. Fast breeder reactors are best suited for India because of
   (a) large uranium deposits (b) large plutonium deposits
   (c) no enriched uranium is required (d) large thorium deposits
17. India's first nuclear power plant was started at
   (a) Kakrapar (Gujarat) (b) Tarapur (Mumbai)
   (c) Kalpakkam (Chennai) (d) Kota (Rajasthan)

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18. Kakrapar nuclear power plant has
(a) fast breeder reactor
(b) pressurised water reactors
(c) CANDU type reactors
(d) boiling water reactors

19. Krypton from gaseous nuclear wastes is removed by
(a) cryogenic treatment
(b) caustic scrubbing
(c) passing through filters
(d) all of the above

Answers

1. (c) **2.** (b) **3.** (a) **4.** (b) **5.** (c) **6.** (a) **7.** (d) **8.** (d) **9.** (b) **10.** (c)
11. (a) **12.** (d) **13.** (a) **14.** (c) **15.** (b) **16.** (d) **17.** (b) **18.** (c) **19.** (a)

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GTU Study Materials ☑

Chapter 11
Jet Propulsion

11.1 Introduction
11.2 Principle of Jet Propulsion
11.3 Classification of propulsive engines
11.4 Ram jet
11.5 Pulse jet engine
11.6 Turbojet
11.7 Thrust, Thrust power, Propulsive efficiency and Thermal efficiency
11.8 Turbojet engine with afterburner
11.9 Turboprop
11.10 Merits and demerits of jet propulsion over other system
11.11 Rocket engine
11.12 Basic theory of operation of rocket engines

11.1 Introduction

Rocket and jet engines are space vehicles. They are propelled by reaction of jet and working fluid coming out with a high velocity. The rocket engines are used in satellites, while jet engines are used in aircraft. Jet engines are also used for cruise missiles and unmanned air vehicles.

11.2 Principle of Jet Propulsion [May '15, Nov. '14]

All the propulsive devices are worked based on two fundamental laws, namely, Newton's Second and Third laws of motion.

Whenever, the momentum is applied to a mass of fluid, a reaction occurs and it gives propulsive force in the opposite direction of fluid flow.

A jet propulsion engine is a reaction engine that discharges a fast moving jet of fluid to generate thrust in the opposite direction of the jet to propel the aircraft. In a jet propulsion engine the propulsive power is not produced by the gas turbine, but it drives the compressor and auxiliary equipment. Hence net work output of a jet propulsion cycle is zero. The exhaust gases that exit in the turbine at relatively higher pressure and temperature, are subsequently passed through nozzle so that the gas with increased velocity in the form of jet comes out into the atmosphere and its reaction in the opposite direction gives the propulsive force to propel the aircraft as shown in Fig 11.1. A propeller discharges the gases at a very high velocity in a large mass of atmospheric air. Since the large mass of atmospheric air cannot be displaced by these gases, thus a reactive force is produced, which propels the aircraft forward (opposite direction of the jet).

The open cycle gas turbine is most suitable for jet propulsion, in which the power developed by turbine is used only to drive the compressor and auxiliary equipments. Then

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further expansion of working fluid takes place in the nozzle which is placed just after the turbine. This unit will serve as a jet propulsion device.

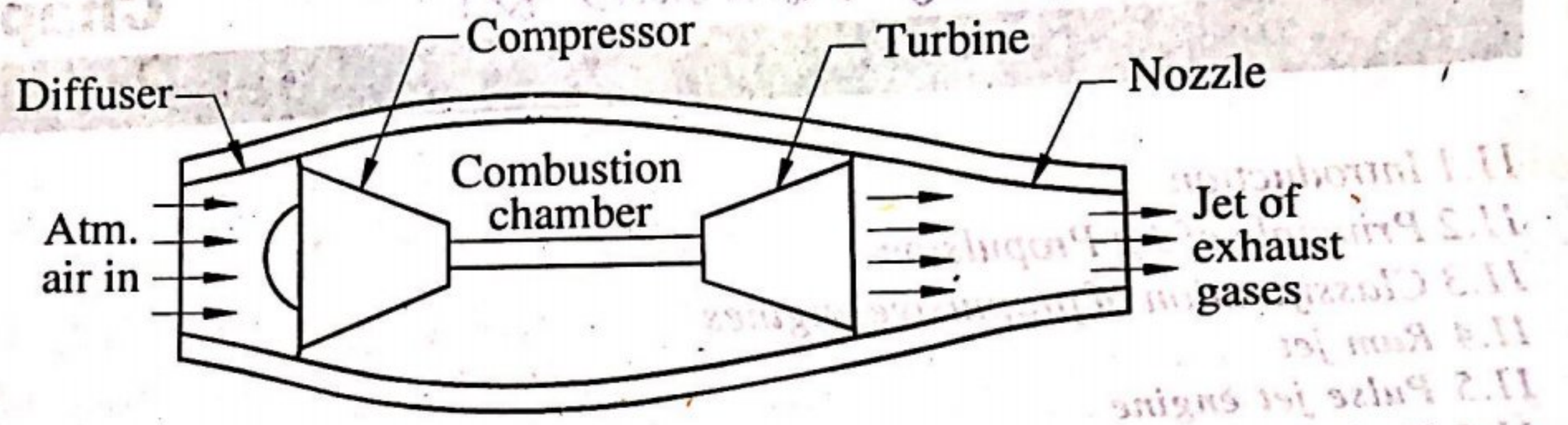
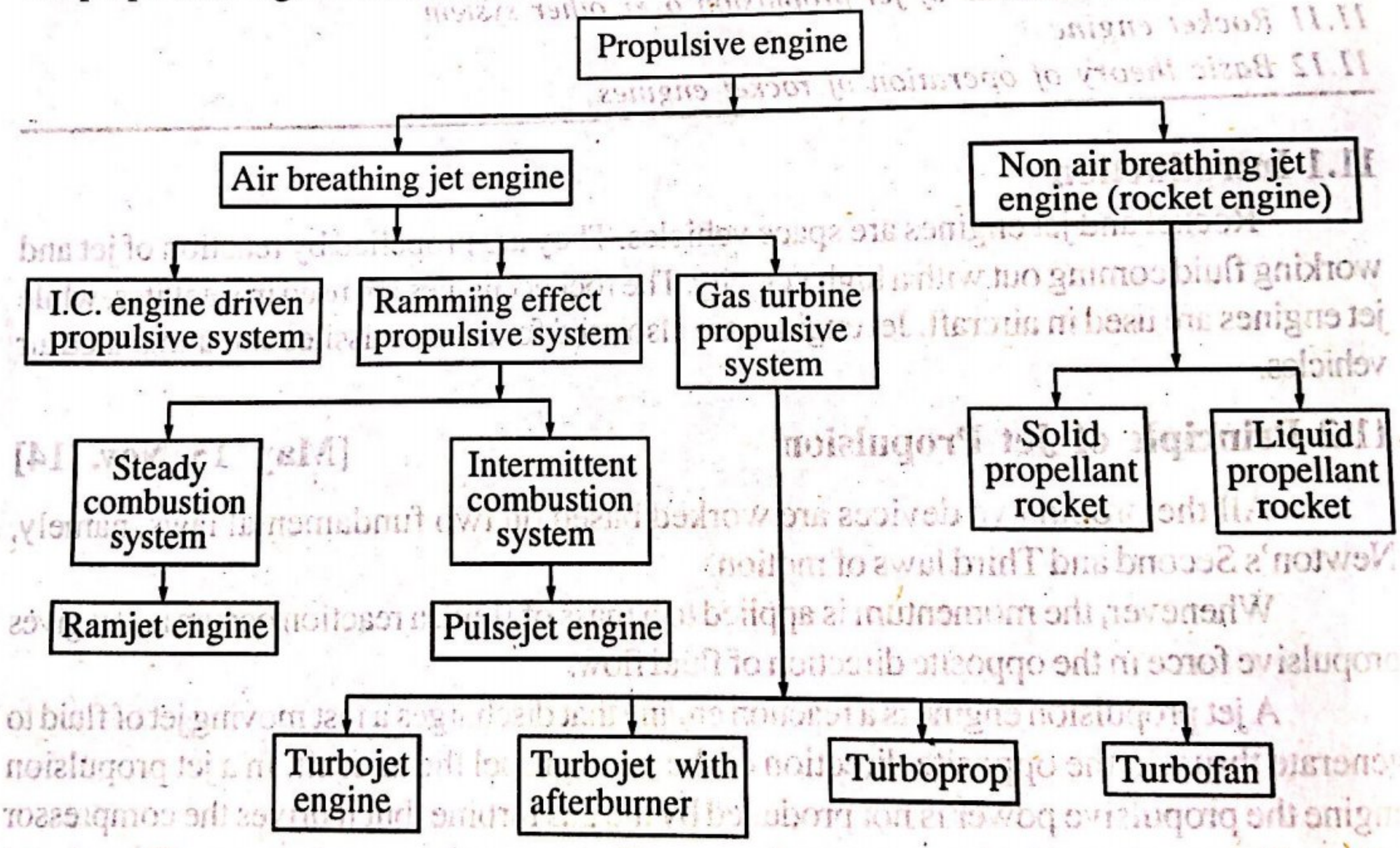


Fig. 11.1 Principle of jet propulsion

11.3 Classification of propulsive engines [May '15]

The propulsive engines can be classified as follows :



Air breathing jet engines : This engine requires oxygen from the surroundings for combustion of the fuel. Here, jet consists of air plus combustion products. The performance of air breathing engines is, therefore, affected by the ambient conditions and the forward speed of the air craft. In this type of engine, the air is taken in, accelerated either by propeller or by expanding high pressure turbine exhaust in a nozzle or by a combination of both. This accelerated mass of combustion product is ejected at the rear of the engine, the resultant change in momentum of the fluid produces a thrust to propel the engine.

Non-air breathing jet (Rocket) engines : In a rocket engine, the fuel is burned in

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the combustion chamber, the source of oxygen being within the rocket and the product of combustion expanded in a nozzle imparting then a high velocity which produces the propulsive force. The performance of non-air breathing (rocket) engines is therefore, not affected by the forward speed and very little affected by ambient condition.

I.C. Engine propulsive system : It is air breathing engine, in which engine takes air as a working fluid. The net power produced by engine is used to drive the propeller which creates thrust as shown in Fig. 11.2. The I.C. engine used in this system in an air cooled, multi-cylinder and radial engine with this engine aircraft can be used for short range flight with a speed limit of 700 km/hr. It is most widely used to power propulsion of helicopters.

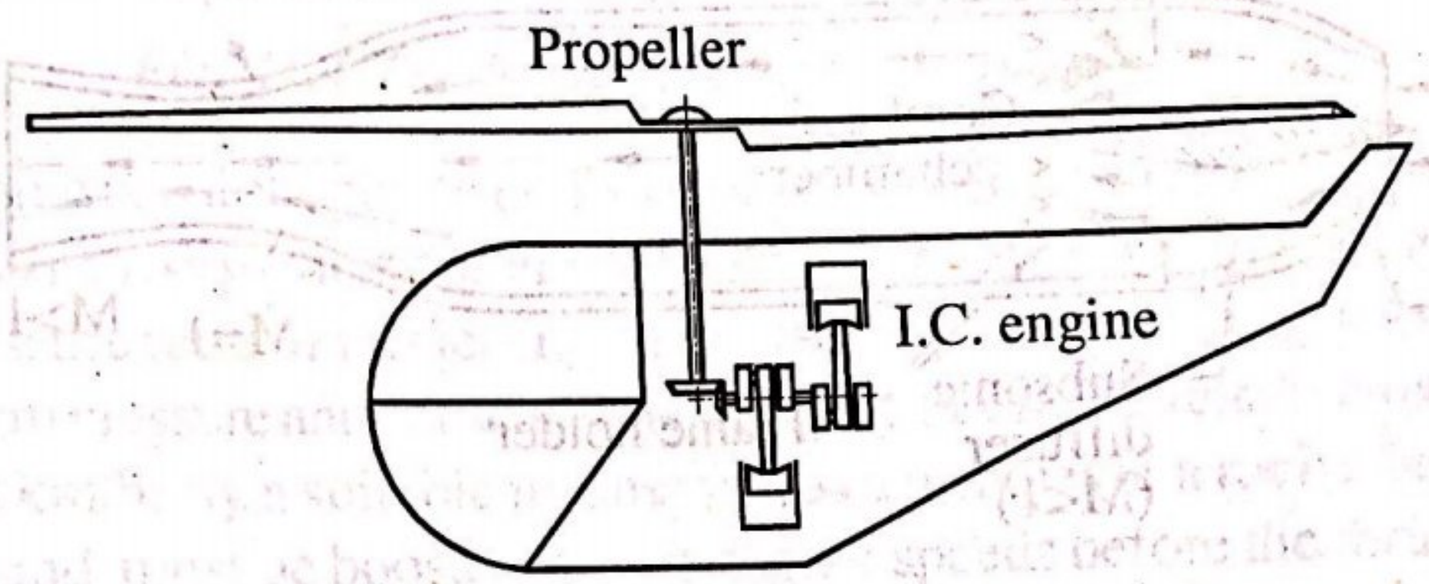


Fig 11.2 I.C. Engine drive propeller engine

- **Difference between Rocket engine and Jet engine**

Rocket engine	Jet engine
(1) Rocket engine is non-air breathing engine.	(1) Jet engine is air breathing engine.
(2) It carries own oxidizer for combustion of fuel. Hence, performance of rocket engine does not depend on ambient conditions and speed of aircraft/rocket.	(2) It takes air from surroundings and this air supplies oxygen for combustion of fuel. Hence, performance of jet engine depends upon ambient conditions and forward speed of aircraft.
(3) Thrust produces by engine does not depend on altitude.	(3) Thrust produced by engine depends on altitude.
(4) There is no surface drag, no gravitational effect. Rate of climb increases with altitude.	(4) There is large skin friction and increases with flight speed.
(5) It can be efficiently operated in vacuum. Space travel is possible.	(5) It cannot be operated in vacuum.
(6) The flight speed may be greater than jet velocity and is unlimited.	(6) The flight speed is always less than jet velocity.
(7) Low thermal efficiency.	(7) Good thermal efficiency.

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11.4 Ram jet [May '15, May '12]

Ram jet is air breathing and steady combustion or continuous flow engine. It was invented by a French engineer Rene Lorin in 1913. It is also called *Athodyd, Larin tube* or *flying stove pipe*.

Construction and working : Ramjet is simplest of all the air breathing engines and consists of supersonic and subsonic diffusers, a combustion chamber with fuel injector and a discharge nozzle as shown in Fig. 11.3.

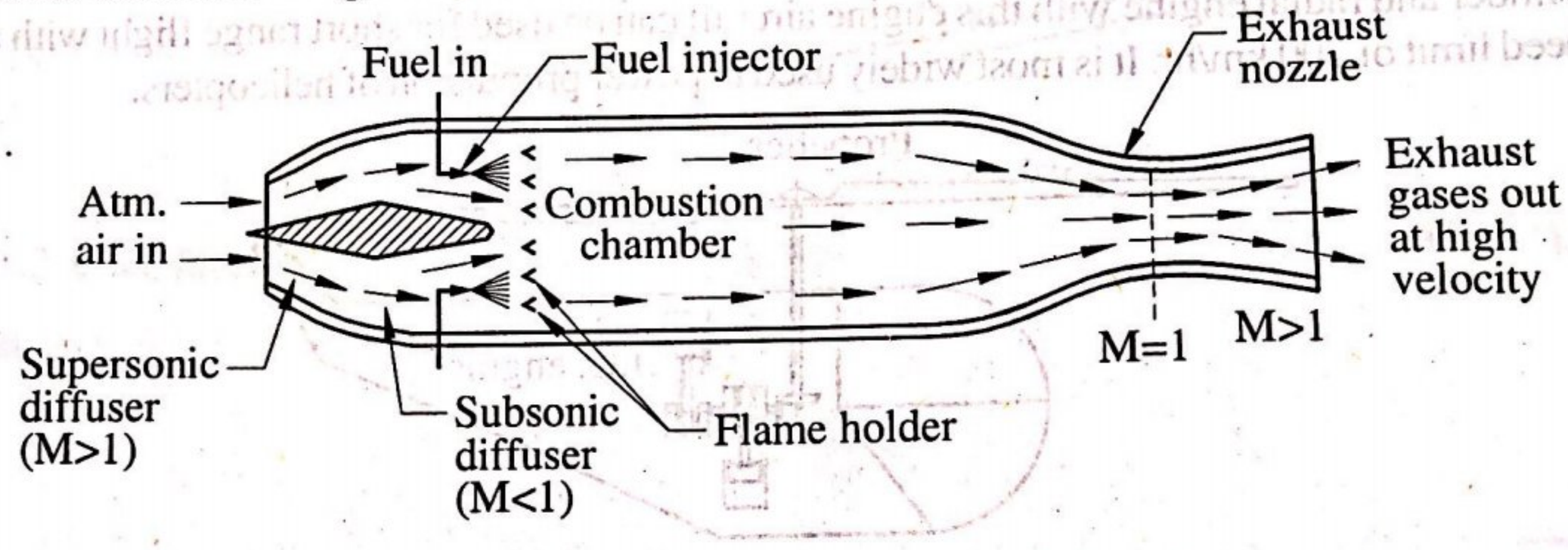


Fig. 11.3 Ramjet engine

Air enters the diffuser at a very high velocity and gets compressed. Air is compressed in two stages by the diffuser. Since usually ramjet operates at supersonic speeds, the first stage of compression takes place in converging section of diffuser. Then second stage of compression takes place in diverging section of diffuser. In diffuser the kinetic energy of air entering is converted into static pressure which is called ram pressure. The air then enters the combustion chamber, fuel is injected where it is ignited by means of a spark plug. The air temperature is raised by continuous combustion of fuel. The hot combustion gases then expand in the exhaust nozzle with a velocity exceeding velocity of air entering. Thus due to change of momentum of working fluid, reaction produced to the ram jet structure through pressure at end of diffuser and shear forces all over the surface of the engine. Thus, thrust is developed in the direction of flight.

The ramjet engine works on Brayton cycle as shown on T-s diagram in Fig. 11.4. In ideal cycle process 1-2 is isentropic ram compression and process 3-4 is isentropic expansion in the nozzle. In actual practice there will be losses due to shock friction and mixing at the diffuser and losses in the nozzle. The actual compression and expansion is shown by the process 1-2′ and 3-4′ respectively. Combustion process is represented by the process 2-3.

Performance of Ramjet : At low speed the fuel consumption in ramjet is high while at high speed it reduces considerably. This is because at low speed the compression is poor. Also, the velocity of ramjet increases due to lower resistance at high altitude. However, thrust produced is reduces as altitude increases due to reduction in density of air with altitude. If increase in thrust by increased velocity is more than decrease in thrust due to reduced density, the specific fuel consumption is reduced.

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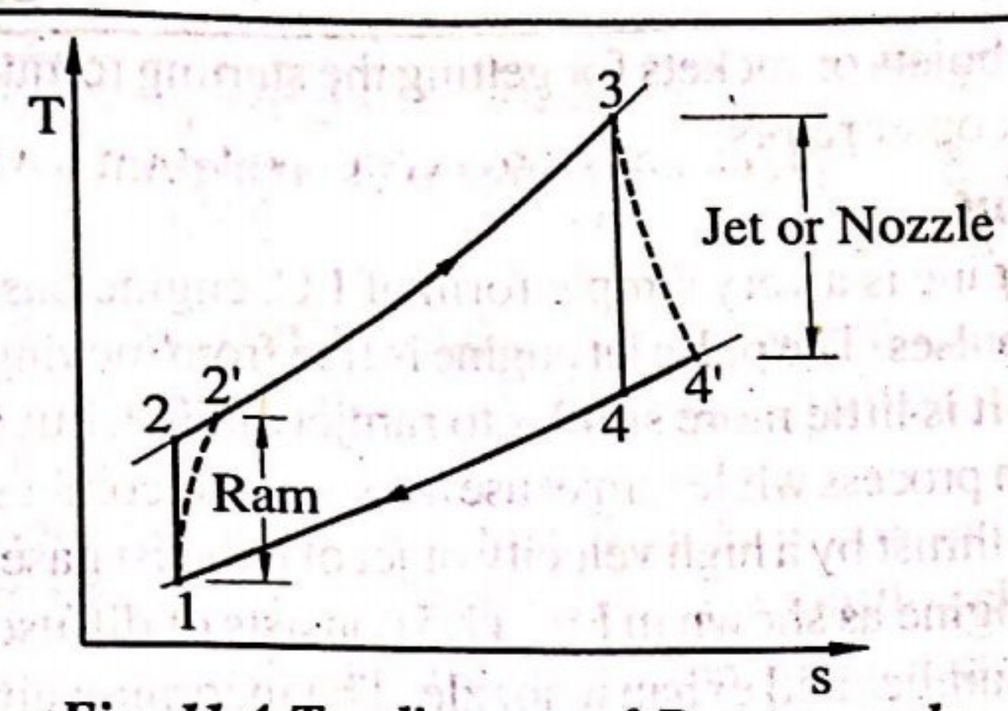


Fig. 11.4 T-s diagram of Brayton cycle

The ramjet engine can not operate under the static condition (zero flight velocity), as there will be no pressure rise in the diffuser and it is not self operating at zero flight velocity. Since, the cycle pressure ratio of ramjet engine depends on its fight velocity. Higher flight velocity, larger the ram pressure and consequently larger the thrust. Therefore ramjet is boosted upto a speed of 290 km/hr by a suitable means such as a turbojet or a rocket before ramjet produce any thrust and must be boosted to even higher speeds before the thrust produced exceeds the drag. The ramjet can be operated at subsonic velocity just below the sonic velocity, it is most efficient at high velocities of about 1700-2200 km/hr (about Mach No. 3) and at very high altitudes.

Advantages of ramjet :

(1) Ramjet is very simple in construction and does not have any moving part such as compressor and turbine.
(2) It is very cheap and requires almost no maintenance.
(3) Greater thrust per unit engine weight than any other propulsion engine at supersonic speed except rocket.
(4) Since, turbine is not used in ramjet, hence maximum temperature of cycle is allowed very high about 2000°C as compared to about 900°C in turbojets. This allows a greater thrust to be obtained by burning fuel at air-fuel ratio of about 15:1.
(5) At very high speed and high altitudes the specific fuel consumption is less than turbojet engine.
(6) It can be operated with any type of liquid fuels and even with solid nuclear fuels.

Disadvantages :

(1) At low and moderate speeds, the fuel comsumption is very high.
(2) It can not be started on its own. It has to be accelerated to a certain flight velocity by some launching device.
(3) Due to high air speed the combustion chamber requires flame holder to stabilize the combustion.

Applications of ramjet : The ramjet engine is widely used in high speed aircraft and missiles due to its high thrust at high operational speed. Subsonic ram jets are used in target weapons,

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in conjuction with turbojets or rockets for getting the starting torque. They have been also used as tip jets on helicopter rotors.

11.5 Pulse jet engine [May '14]

A pulse jet engine is a very simple form of I.C. engine based jet engine in which combustion occurs in pulses. The pulse jet engine is free from moving parts like compressor, turbine and propeller. It is little more similar to ramjet engine, but pulse jet engine use on intermittent combustion process while ramjet use a continuous combustion process. The pulse jet engine develops the thrust by a high velocity of jet of exhaust gases.

The pulse jet engine as shown in Fig. 11.5 consists of diffuser, one way flap or reed valve, a combustion chamber and exhaust nozzle. The incoming air is compressed by ram effect in the diffuser section and then passes through the passages which are connected with diffuser and opened or closed by reed valves. Fuel in the form of a gas or liquid aerosol is either mixed with the air in the intake or injected into the combustion chamber. The combustion is then initiated by a spark plug. As combustion takes place the tempressure and pressure of combustion gases became higher than the ram pressure, hence the reed valves get closed. The high pressure hot gases then exit through the exhaust nozzle with high kinetic energy and produce forward thrust on the unit. Due to escape of gases to the atmosphere the pressure reduces in the combustion chamber, therefore the ram air forces the reed valve to open and fresh air enters in the combustion chamber for the next cycle.

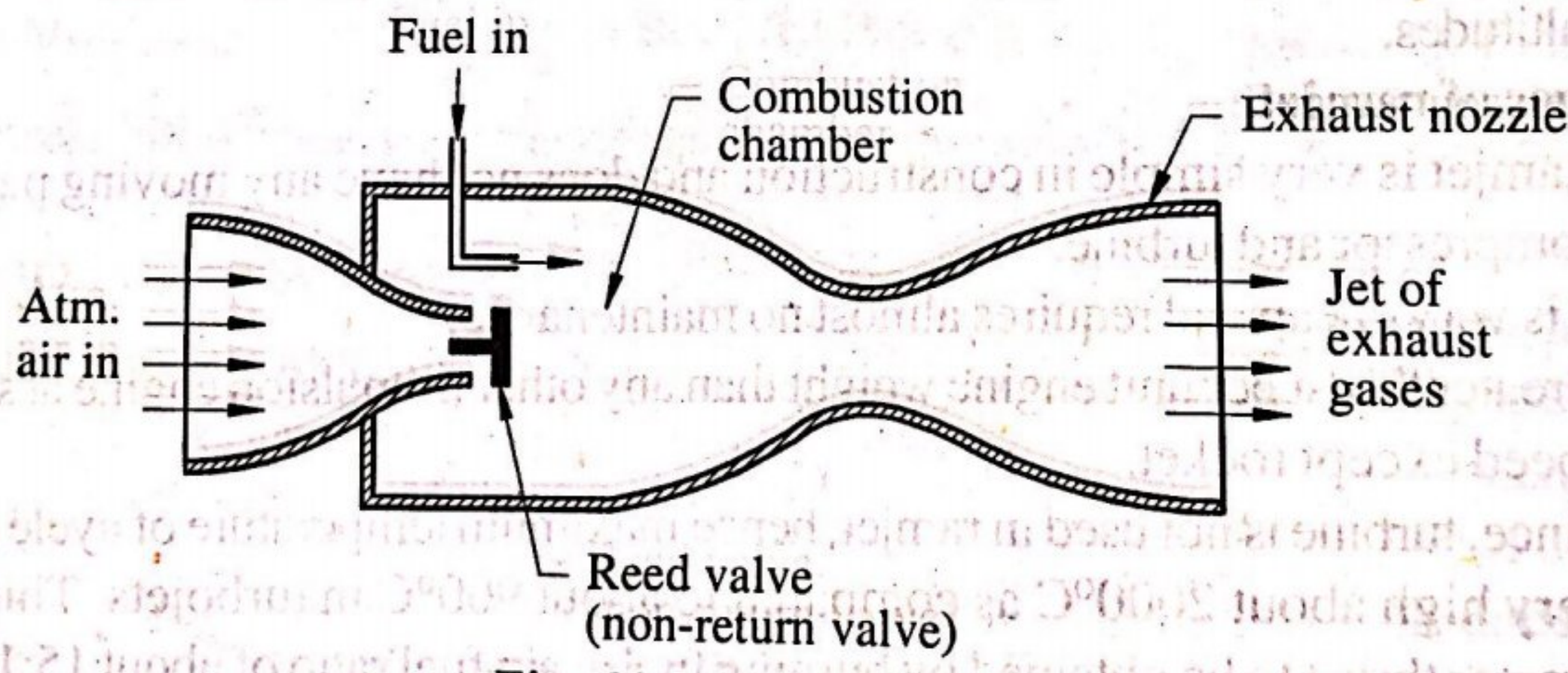


Fig. 11.5 Pulse jet engine

Advantages of pulse jet engine :

(1) It is simple in construction compared to turbojet engine.
(2) It does not having moving parts such as compressor, turbine and propeller. Hence it is light, and can carry high pay load.
(3) It can be produced sufficient thrust at low speeds.
(4) It is highly suitable for bombers and target missiles because it can be operated without pilot.
(5) It is cheaper then turbojet engine.

Disadvantages :

(1) It produces very high noise and severe vibrations due to intermittent pulse.
(2) It has high fuel consumption and very low thermal efficiency (about 2 to 3%).

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Process 0-1 : Ram compression in diffuser

Air enters the diffuser from the surrounding atmosphere at temperature T_0 and pressure p_0 at flight velocity C_a. The air is slowed down in the diffuser and kinetic energy of air is converted into static pressure. The pressure, temperature and velocity distribution is given in Fig. 11.8.

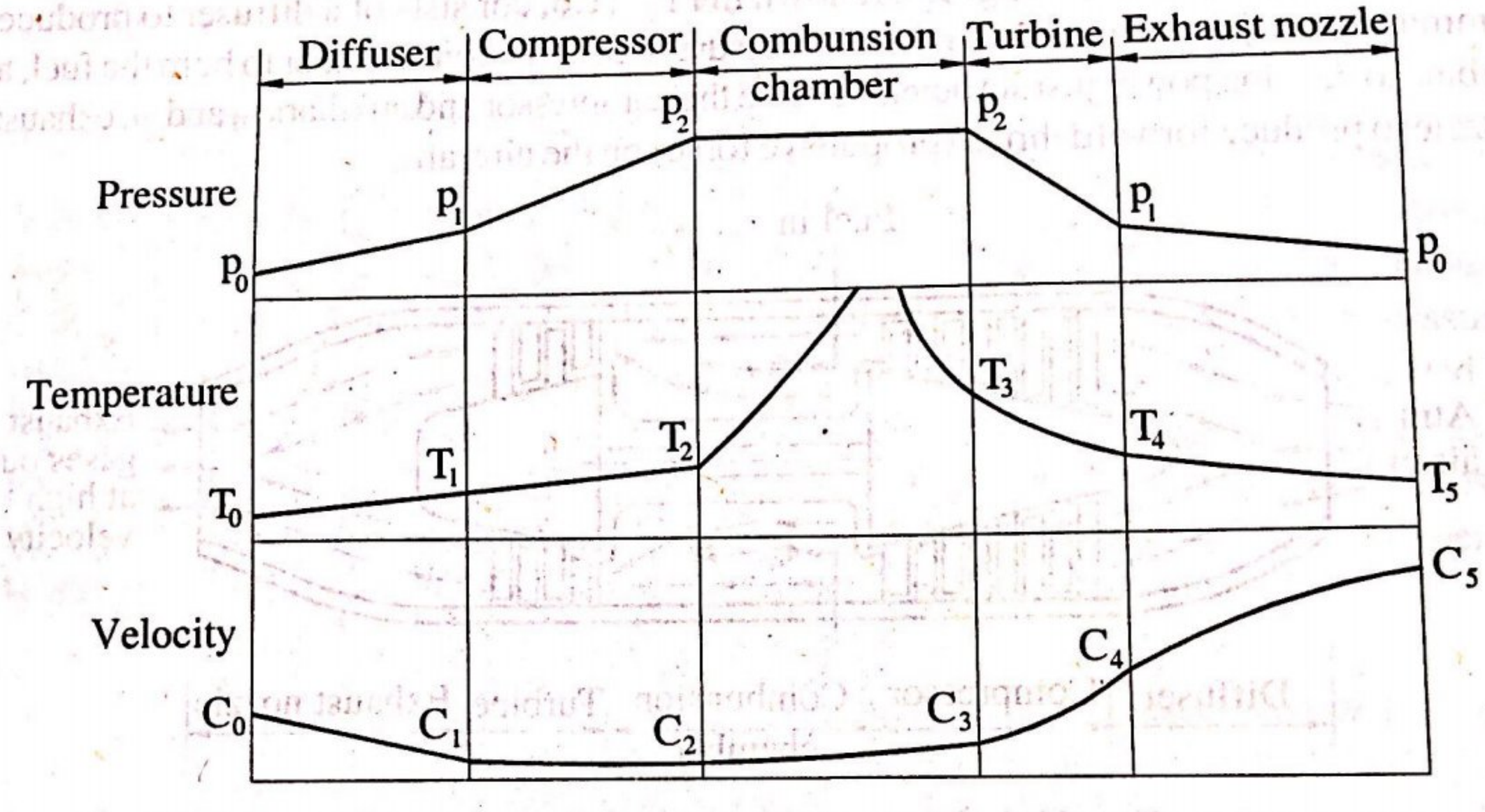


Fig. 11.8 pressure, temperature and velocity distribution in turbojet engine.

Energy equation between state 0 and 1.

$$\frac{C_a^2}{2} + h_0 + q_{01} = \frac{C_1^2}{2} + h_1 + w_{01}$$

For ideal diffuser, $C_1 = 0$, $q_{01} = 0$, , and $w_{01} = 0$

$$\therefore h_1 = h_0 + \frac{C_a^2}{2} \text{ kJ/kg} \qquad ...(11.1)$$

$$\therefore T_1 = T_0 + \frac{C_a^2}{2C_p} \qquad ..(11.2)$$

$$\text{Diffuser efficiency } \eta_d = \frac{\text{Ideal (isentropic) enthalpy rise}}{\text{Actual enthalpy rise}}$$

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$$= \frac{h_1 - h_0}{h_1' - h_0} = \frac{T_1 - T_0}{T_1' - T_0}$$

$\therefore h_1 - h_0 = \eta_d (h_1' - h_0)$ and $T_1 - T_0 = \eta_d (T_1' - T_0)$

Hence equation (11.1) and (11.2) becomes

$$h_1' = h_0 + \frac{C_a^2}{2\eta_d} \quad ...(11.3)$$

and $$T_1' = T_0 + \frac{C_a^2}{2C_p\eta_d} \quad ...(11.4)$$

Process 1-2 : Isentropic compression in compressor

The diffused air at pressure p_1 and temperature T_1 enters the compressor and is compressed to pressure p_2.

From energy equation we get, ideal work consumed by compressor

$$w_c = h_2 - h_1 = C_p (T_2 - T_1) \quad ...(11.5)$$

Compressor efficiency $\eta_c = \frac{h_2 - h_1}{h_2' - h_1} = \frac{T_2 - T_1}{T_2' - T_1}$

$\therefore$ Actual compressor work $w_c = h_2' - h_1 = C_p (T_2' - T_1)$

$$\therefore w_c = \frac{h_2 - h_1}{\eta_c} = \frac{C_p (T_2 - T_1)}{\eta_c} \quad ...(11.6)$$

Process 2-3 : Constant pressure heat addition in combustion chamber

Heat supplied to the combustion chamber

$$Q_s = (m_a + m_f) h_3 - m_a h_2 \quad ...(11.7)$$

where m_a = mass flow rate of air

m_f = mass flow rate of fuel supplied

Combustion efficiency is given by

$$\eta_{comb} = \frac{\text{Actual rise in enthalpy of gases}}{\text{Energy supplied by fuel}} = \frac{(m_a + m_f) h_3 - m_a h_2}{m_f \cdot CV}$$

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Process 3-4 : Isentropic expansion in turbine

The hot combustion gases generated in the combustion chamber expands partially in the turbine from pressure p_3 and temperature T_3 to pressure p_4 and temperature T_4.

From energy equation we get, isentropic work produced by turbine

$$w_t = h_3 - h_4 = C_p(T_3 - T_4) \quad ...(11.8)$$

Isentropic efficiency of turbine $\eta_t = \dfrac{h_3 - h_4'}{h_3 - h_4} = \dfrac{T_3 - T_4'}{T_3 - T_4}$

Actual work output of turbine $w_t = h_3 - h_4' = C_p(T_3 - T_4')$

$$\therefore w_t = (h_3 - h_4)\eta_t = C_p(T_3 - T_4)\eta_t \quad ...(11.9)$$

Theoretically, for turbojet Compressor work input = Turbine work output

$$\therefore \frac{C_p(T_2 - T_1)}{\eta_c} = C_p(T_3 - T_4)\eta_t$$

$$\therefore T_2 - T_1 = (T_3 - T_4)\eta_c \cdot \eta_t \quad ...(11.10)$$

Process 4-5 : Isentropic expansion in nozzle

After partial expansion of combustion gases in the turbine, the gases enter the nozzle and expand isentropically from pressure p_4 and temperature T_4 to pressure p_5 and temperature T_5.

Energy equation between state 4 and 5,

$$\frac{C_4^2}{2} + h_4' = \frac{C_5^2}{2} + h_5 \text{ (Ideal)} \quad ...(11.11)$$

$$\frac{C_4'^2}{2} + h_4' = h_5' + \frac{C_5'^2}{2} \text{ (actual)} \quad ...(11.12)$$

If C_4' is very less compared to $C_5' = C_j$ = jet velocity, then neglecting C_4' we get,

$$h_4' = h_5' + \frac{C_j^2}{2} \quad ...(11.13)$$

Nozzle efficiency is given by $\eta_n = \dfrac{\text{Actual enthalpy drop}}{\text{Isentropic enthalpy drop}} = \dfrac{h_4' - h_5'}{h_4' - h_5}$

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$$\therefore h_4' - h_5' = (h_4' - h_5)\eta_n$$

Hence equation (8.13) becomes

$$h_4' = h_5 + \frac{C_j^2}{2\eta_n}$$

$$\therefore C_p T_4' = C_p T_5 + \frac{C_j^2}{2\eta_n}$$

$$\therefore \boxed{C_j = \sqrt{2\eta_n \cdot C_p (T_4' - T_5)}} \quad ...(11.14)$$

11.7 Thrust, Thrust power, Propulsive efficiency and Thermal efficiency

[May '14]

Thrust (T) : It is unbalance force acting on space vehicles due to difference in momentum of flow velocity of air entering the engine and high velocity hot gases leaving the exhaust nozzle. According to Newton's second law of motion

$$\text{Thrust, } T = (m_a + m_f)C_j - m_a C_a, \text{ Newton}$$

where, C_j = velocity of gases leaving the nozzle

C_a = velocity of air entering the engine

$$\therefore \boxed{T = \left(1 + \frac{m_f}{m_a}\right) C_j - C_a \text{ N/kg of air/s}} \quad ...(11.15)$$

Since, the air-fuel ratio used in the aircraft is very high. the mass flow rate of exhaust gases almost same as of air entering the engine. Neglecting the mass of fuel.

$$T = C_j - C_a, \text{ N/kg of air/s} \quad ...(11.16)$$

OR $\boxed{T = m_a (C_j - C_a), \text{ N}}$

Thrust power (T.P.) : It is the power developed from thrust of engine. It is the product of thrust and the velocity of vehicle.

$$\text{Thrust power, } \boxed{T \cdot P = \left[\left(1 + \frac{m_f}{m_a}\right) C_j - C_a\right] \times C_a, \text{ W/kg of air}} \quad ...(11.17)$$

Neglecting mass of fuel, we get

$$\boxed{T \cdot P = (C_j - C_a) \times C_a, \text{ W/kg of air}} \quad ...(11.18)$$

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OR $$T \cdot P = \frac{(C_j - C_a) \times C_a}{1000}, \text{kW/kg of air}$$

Propulsive power (PP) : It is energy required to change the momentum of the mass flow of gas. It is expressed as the difference between the rate of kinetic energies of the entering air and gas leaving the nozzle.

$$\text{P.P.} = \Delta\text{KE} = (m_a + m_f)\frac{C_j^2}{2} - m_a \frac{C_a^2}{2}, \text{Watt}$$

$$\therefore \text{PP} = \left(1 + \frac{m_f}{m_a}\right)\frac{C_j^2}{2} - \frac{C_a^2}{2}, \text{W/kg of air} \quad \ldots(11.19)$$

Neglecting mass of fuel we get

$$\text{P.P.} = \frac{C_j^2 - C_a^2}{2}, \text{W/kg of air} \quad \ldots(11.20)$$

$$\therefore \text{P.P} = \frac{C_j^2 - C_a^2}{2 \times 1000}, \text{kW/kg of air}$$

Propulsive efficiency (η_p) : It is ratio of thrust power produced to propulsive power required.

$$\eta_p = \frac{\text{Thrust power}}{\text{Propulsive power}} = \frac{\text{TP}}{\text{PP}}$$

$$\therefore \eta_p = \frac{\left[\left(1 + \frac{m_f}{m_a}\right)C_j - C_a\right] \times C_a}{\left(1 + \frac{m_f}{m_a}\right)\frac{C_j^2}{2} - \frac{C_a^2}{2}} \quad \ldots(11.21)$$

Neglecting mass of fuel, we get

$$\eta_p = \frac{(C_j - C_a) \times C_a}{\frac{1}{2}(C_j^2 - C_a^2)} = \frac{2C_a(C_j - C_a)}{(C_j - C_a)(C_j + C_a)}$$

$$\therefore \eta_p = \frac{2C_a}{C_j + C_a} = \frac{2}{(C_j / C_a) + 1} \quad \ldots(11.22)$$

From above equation (11.22), it is clear that propulsive efficiency increases as the aircraft

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speed (C_a) approaches jet velocity (C_j), but thrust decreases.

Thermal efficiency of the plant (η_{th}) : It is the ratio of propulsive power to rate heat supply to the system.

$$\eta_{th} = \frac{\text{Propulsive power}}{\text{Rate of heat supply}} = \frac{\text{P.P.}}{Q_s}$$

$$\eta_{th} = \frac{\left(1+\frac{m_f}{m_a}\right)\frac{C_j^2}{2} - \frac{C_a^2}{2}}{\frac{m_f}{m_a}\times CV} \quad \ldots(11.23)$$

Neglecting mass of the fuel we get

$$\eta_{th} = \frac{m_a(C_j^2 - C_a^2)}{2m_f \cdot CV} \quad \ldots(11.24)$$

Overall efficiency of plant ($\eta_{overall}$) : It is a product of thermal efficiency and propulsive efficiency. It can be defined as ratio of thrust power to rate of heat supplied to the system.

$$\eta_{overall} = \frac{\text{Thrust power}}{\text{Rate of heat supply}} = \frac{\text{T.P.}}{Q_s}$$

$$\therefore \eta_{overall} = \frac{\left[\left(1+\frac{m_f}{m_a}\right)C_j - C_a\right]\times C_a}{\frac{m_f}{m_a}\times CV} \quad \ldots(11.25)$$

Neglecting mass of fuel we get

$$\eta_{overall} = \frac{m_a(C_j - C_a)\times C_a}{m_f \times CV} \quad \ldots(11.26)$$

Rearranging equation (11.25), we get

$$\eta_{overall} = \frac{\left(1+\frac{m_f}{m_a}\right)\frac{C_j^2}{2} - \frac{C_a^2}{2}}{\frac{m_f}{m_a}\times C.V.} \times \frac{\left[\left(1+\frac{m_f}{m_a}\right)C_j - C_a\right]\times C_a}{\left(1+\frac{m_f}{m_a}\right)\frac{C_j^2}{2} - \frac{C_a^2}{2}}$$

$$\therefore \eta_{overall} = \eta_{th} \times \eta_p$$

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Thrust specific fuel consumption : It is ratio of fuel flow rate to thrust produced.

$$\text{Thrust sfc} = \frac{\text{Fuel flow rate}}{\text{Thrust produced}} = \frac{m_f}{m_a(C_j - C_a)} \quad ...(11.27)$$

Advantages of Turbojet engine :

(1) It is simple in construction as compared to I.C. engine propelled system for same power output.
(2) The weight power ratio of turbojet engine is one fourth that of a I.C. engine propelled system.
(3) The speed of turbojet is not limited by the propeller and it can attain higher flight speeds than I.C. Engine propeller aircrafts.
(4) The thrust or power output of turbojet is not limited.
(5) The turbojet is very suitable for longer flights at higher altitude and speed.
(6) Less maintenance is required.

Disadvantages

(1) The fuel economy at low flight speeds is extremely poor.
(2) Propulsive efficiency and thrust are lower at low speed. It becomes inefficient below a speed of 550 km/hr.
(3) Turbojet is not economical for short distance flights. Also it requires longer runway for it take off and landing.
(4) High capital cost.

11.8 Turbojet engine with afterburner

As we know that turbojet has a poor performance characteristics at low speed and take-off. Several modification to turbojet engine have been made to fulfill the special performance requirement of various aircraft. The thrust from a turbojet is given by

$$T = (m_a + m_f)C_j - m_a C_a$$

From above equation it is clear that the thrust can be increased by increasing velocity of exhaust gases (C_j). The C_j is the function of maximum temperature in the cycle, as maximum temperature increases, the exit gases velocity C_j also increases, and hence thrust increases. The thrust can also be increased by increasing the mass flow rate of gases.

An afterburner is a device added to the rear of the turbojet engine as shown in Fig. 11.9, which is one of the method to increase the thrust of turbojet engine for short duration for better take-off performance, higher rate of climb and increased performance at higher altitude. The exhaust of the turbine of turbojet has sufficient oxygen to burn as there is always excess air available. This allows the burning before it reaches the nozzle. It is called ***after burning***. This increases the mass flow and the velocity of the exhaust jet, hence thrust of aircraft is increased. Its purpose is to provide a temporary increase in thrust, both for supersonic flight and for take-off. 30% of thrust can be increased with the help of afterburner. However, the fuel consumption

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increases rapidly. In order to increase 20% of thrust, the 100 % fuel consumption increases. Hence, afterburner is used only for take-off or for high climbing rates and for a very short duration. Also, exhaust jet area should be variable to meet the demands of increased flow rate.

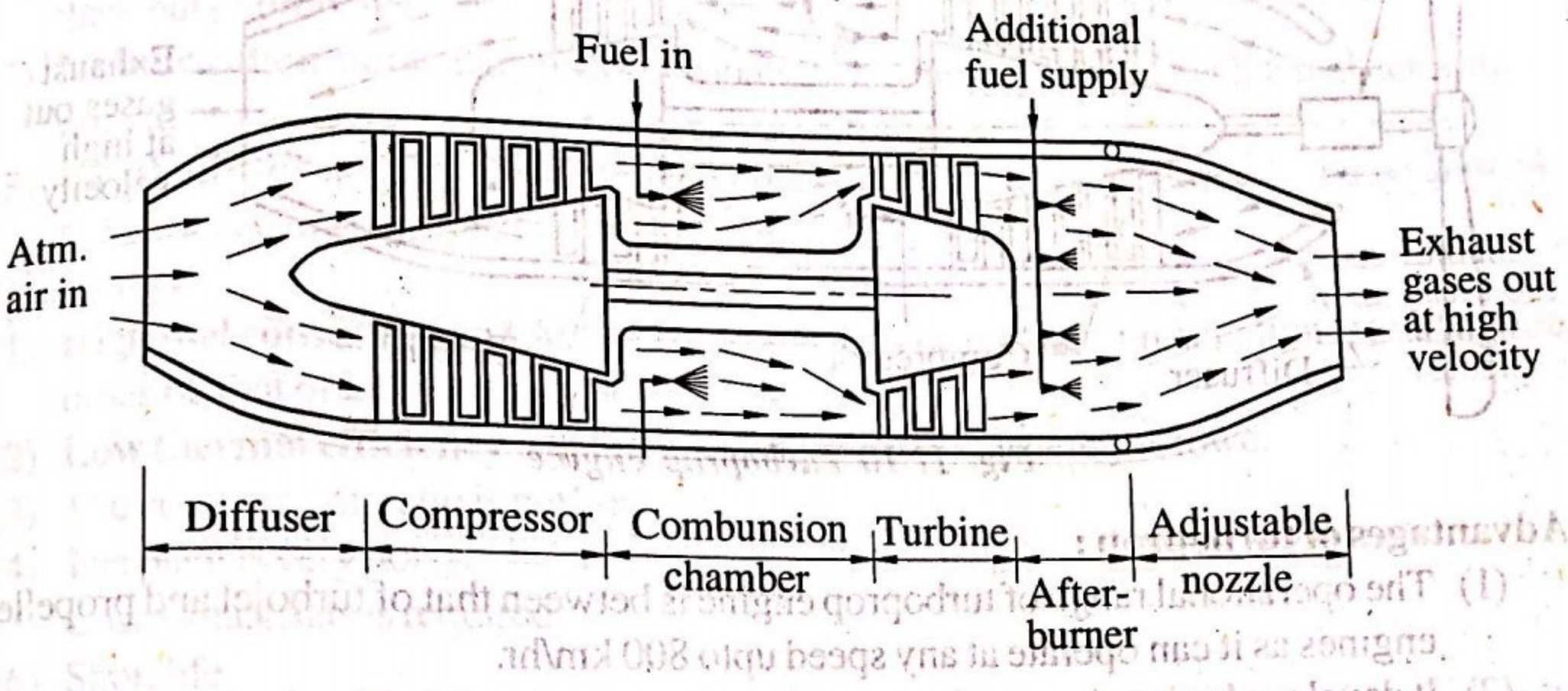


Fig. 11.9 Turbojet engine with afterburner

11.9 Turboprop [May '12]

As we know that the turbojet engine or other jet engine has a poor performance at low speed and take-off conditions, while I.C. engine propelled system is used only for short range flight with speed limited of 700 km/hr. A higher thrust per unit mass flow of fuel can be obtained by increasing the mass flow of air and that this also results in better fuel economy. This fact is utilized in turboprop engine which is an intermidiate device between a jet engine and propeller driven by I.C. engine. A turboprop engine is also called ***propjet***.

A turboprop engine consists of diffuser, compressor, combustion chamber, turbine, exhaust nozzle and propeller coupled with turbine shaft and driven by turbine as shown in Fig. 11.10. The air flow increases by propeller in addition to the thrust produced by the exhaust nozzle.

The air enters the diffuser from the surroundings where kinetic energy of air reduces and pressure increased. Then air is compressed in compressor before passing to the combustion chamber. The compressor in the turboprop is essentially an axial flow compressor. The product of combustion expand in a two-stage turbine. One stage of the turbine drives the compressor and the other drives the propeller. Hence, less energy available for expansion in the nozzle. About 80% of the available energy in gases is utilized to run compressor and propeller while rest about 20% energy contributes the thrust by velocity difference of gases leaving and air entering the aircraft. The propeller also provides the usefull thrust.

The reduction gear box is used between propeller and turbine shafts to reduce rpm at propeller shaft.

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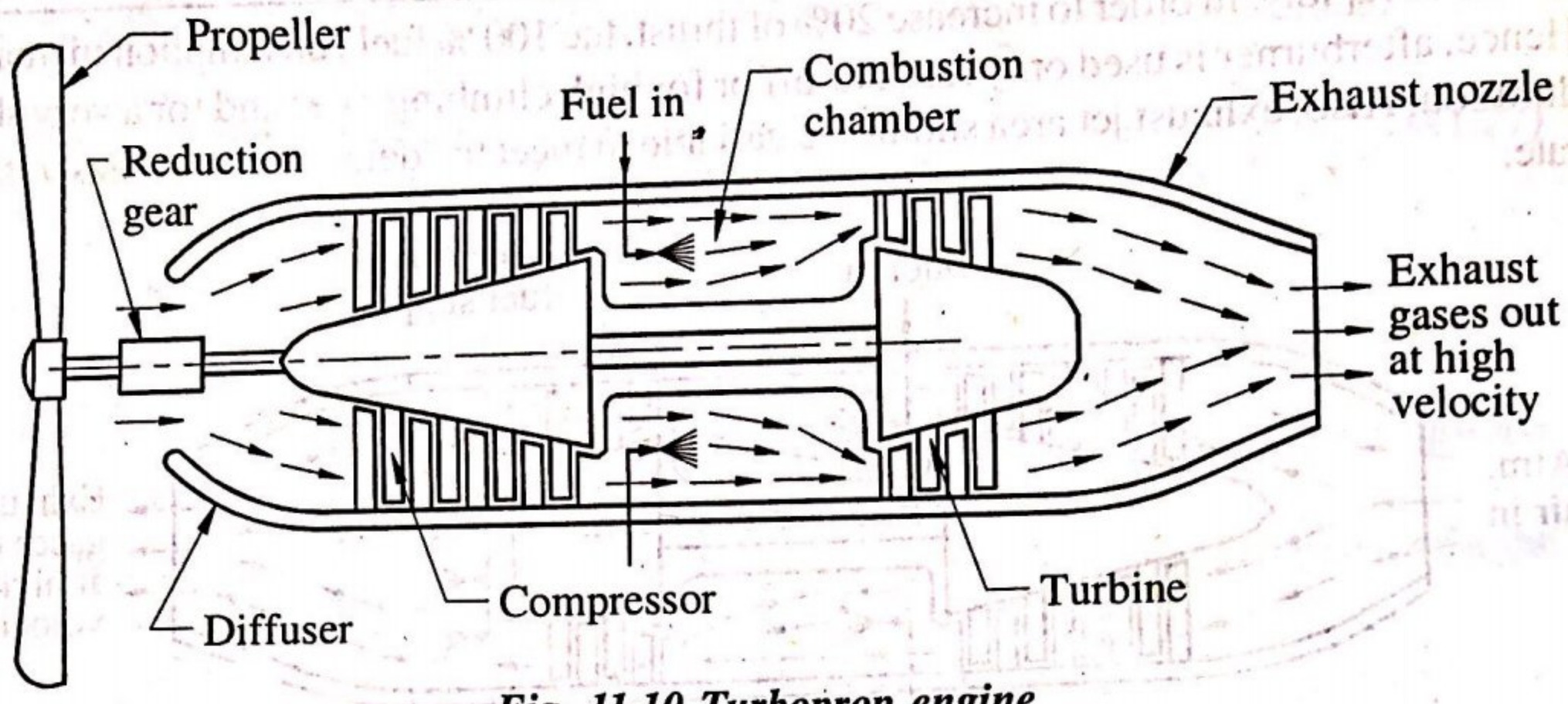


Fig. 11.10 Turboprop engine

Advantages of turboprop :

(1) The operational range of turboprop engine is between that of turbojet and propeller engines as it can operate at any speed upto 800 km/hr.

(2) It develops higher thrust at low speeds and hence take off rolling in short, requiring shorter runway. It is very useful for short range flights with a speed of 600 km/hr.

(3) The turboprop engines are utilized the advantages of both the pure jet engine and propeller engine as low specific weight, small frontal area, high speed, large thrust compared to propelled engine and high take-off thrust, good propeller efficiency at lower speeds compared to pure jet engine.

(4) Thrust reversal is easily achieved by varying the blade angle, and aircraft speed can be drastically decreased.

Disadvantages

(1) At high speed, the propulsive efficiency decreases drastically, thereby, putting up a maximum speed limit on the engine.

(2) It is used only for short range of flights, low altitude and low speeds.

(3) It requires reduction gear which increases the cost and weight of aircraft.

11.10 Merits and demerits of jet propulsion over other system [Nov.'13]

Merits :

(1) **Small frontal area :** The frontal area of jet propulsion engine is less than one-forth of that of I.C. engine propelled engine.

(2) **Low specific weight :** The specific weight of jet propulsion engine is from one-fourth to one-half of that of I.C. engine propelled engine.

(3) **No unbalance force :** Jet propulsion engine is free from unbalance forces because there is no reciprocating parts.

(4) **High Speed :** The speed of I.C. engine driven aircraft is limited to 800 km/hr while speed

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of jet propulsion engine is not limited.

(5) **High power output or thrust :** The power output of I.C. engine propelled system is limited due to detonation while power output of jet propulsion engine is not limited.

(6) **High efficiency :** At high altitude (above 10,000 m) the efficiency of jet is much higher than that of propeller.

(7) **No lubrication nor radiators :** Jet propulsion engine requires neither internal lubrication nor radiators as it requires in I.C. engine driven engine.

(8) **Flexibility for position of pilot :** Jet propulsion engine aircraft permits better position of the pilot due to no propeller at from of aircraft.

Demerits :

(1) **High fuel consumption :** At low speed and altitude the fuel consumption is two to three times of that of I.C. engine propelled aircrafts.

(2) **Low thermal efficiency :** At low pressure thermal efficiency is lower.

(3) The compression ratio is not constant.

(4) The plant is very noisy.

(5) Costly materials is required.

(6) Short life.

Problem 11.1 *A turbojet engine having two jets, takes air at velocity 200 m/s when flying at an altitude of 10,000 m. The resistance or drag of the plane is 6500 N. The air fuel ratio is 50:1. The propulsive efficiency of jet is 50%. Calculate (i) absolute velocity of jet, (ii) mass flow rate of air enters, (iii) propulsive power, (iv) mass of fuel supplied and (v) thrust specific fuel consumption.*

Solution : Given data : $C_a = 200$ m/s, $n_{jet} = 2$, T = 6500 N, $m_a / m_f = 50:1$, $\eta_p = 50\%$,

C.V. = 48,000 kJ/kg, $C_j = ?$, $m_a = ?$, PP = ?, $m_f = ?$, Thrust sfc = ?

(i) Absolute velocity of jet $(C_j - C_a)$

Propulsive efficiency is given by $\eta_p = \dfrac{\left[\left(1+\dfrac{m_f}{m_a}\right)C_j - C_a\right]\times C_a}{\left(1+\dfrac{m_f}{m_a}\right)\dfrac{C_j^2}{2} - \dfrac{C_a^2}{2}}$

$$\therefore\ 0.5 = \frac{\left[\left(1+\frac{1}{50}\right)C_j - 200\right]\times 200}{\left(1+\frac{1}{50}\right)\frac{C_j^2}{2} - \frac{(200)^2}{2}}$$

$$\therefore\ 0.51C_j^2 - 20{,}000 = 408C_j - 80{,}000$$

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$$\therefore 0.51C_j^2 - 408C_j + 60,000 = 0$$

Solving above equation, we get

$$\therefore C_j = 605.8 \ m/s$$

$\therefore$ Absolute velocity of gases $= C_j - C_a = 605.8 - 200 = \mathbf{405.8\ m/s}$ ***Ans***

(ii) Mass flow rate of air (m_a)

$$\text{Thrust } T = (m_a + m_f)C_j - m_a C_a$$

$$\therefore 6500 = m_a\left[\left(1+\frac{1}{50}\right)\times 605.8 - 200\right]$$

$\therefore m_a = \mathbf{15.55\ kg/s}$ ***Ans***

(iii) Propulsive power (PP)

$$\text{PP} = (m_a + m_f)\frac{C_j^2}{2} - m_a\frac{C_a^2}{2}$$

$$= \frac{m_a}{2}\left[\left(1+\frac{m_f}{m_a}\right)C_j^2 - C_a^2\right]$$

$$= \frac{15.55}{2}\times\left[\left(1+\frac{1}{50}\right)\times(605.8)^2 - (200)^2\right]$$

$= 2599443\ \text{W} = \mathbf{2599.44\ kW}$ ***Ans***

(iv) Mass of fuel supplied (m_f)

$$\text{Air fuel ratio} = \frac{m_a}{m_f} = 50$$

$$\therefore m_f = \frac{15.55}{50} = \mathbf{0.311\ kg/s}$$ ***Ans***

(v) Thrust specific fuel consumption

$$\text{Thrust sfc} = \frac{m_f}{T} = \frac{0.311}{6500} = \mathbf{4.78\times 10^{-5}\ kg/Ns}$$ ***Ans***

Problem 11.2 *A turbojet engine aircraft flies with a velocity of 250 m/s at an altitude where the air is a 0.32 bar and –35°C. The air enters the compressor at a rate of 10 kg/s. The pressure ratio in the compressor is 10 and temperature at inlet of turbine is*

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1000°C. Assume isentropic compression in diffuser and compressor and isentropic expansion in turbine and nozzle. For air, C_p = 1005 J/kg K, γ = 1.4. Calculate (i) velocity of gases at the nozzle exit, (ii) propulsive efficiency of cycle and (iii) thermal efficiency.

Solution : Given data : $C_a = 250$ m/s, $p_1 = 0.32$ bar, $T_0 = -35°C = 238$ K,

$m_a = 10$ kg/s, $p_2 / p_1 = 10$, $T_3 = 1000°C = 1273$ K, $C_j = ?$, $\eta_p = ?$

For process 0-1 : Isentropic compression in diffuser (Refer Fig. 11.11)

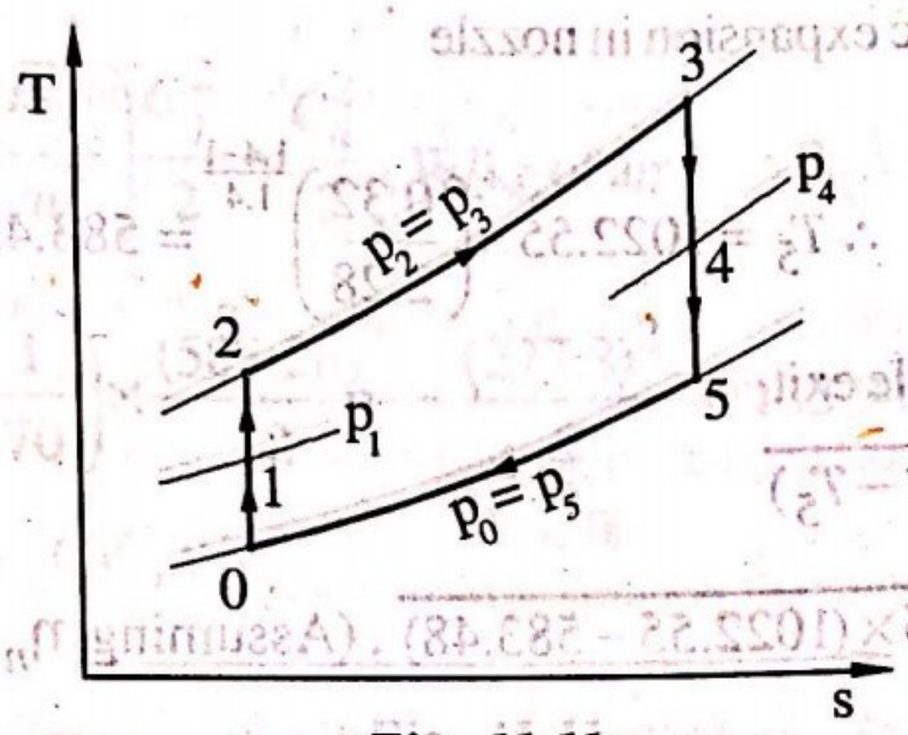


Fig. 11.11

For ideal diffuser, $T_1 = T_0 + \frac{C_a^2}{2C_p} = 238 + \frac{(250)^2}{2\times1005} = 269.09$ K

Also, $\frac{T_1}{T_0} = \left(\frac{p_1}{p_0}\right)^{\frac{\gamma-1}{\gamma}}$ $\therefore p_1 = p_0\left(\frac{T_1}{T_0}\right)^{\frac{\gamma}{\gamma-1}}$

$$\therefore p_1 = 0.32\times\left(\frac{269.09}{238}\right)^{\frac{1.4}{1.4-1}} = 0.492 \text{ bar}$$

For process 1-2 : Isentropic compression in compressor

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \quad \therefore T_2 = 269.09(10)^{\frac{1.4}{1.4-1}} = 519.53 \text{ K}$$

For turbojet, Compressor input = Turbine work output

$$\therefore C_p(T_2 - T_1) = C_p(T_3 - T_4)$$

$$\therefore T_4 = T_3 - (T_2 - T_1) = 1273 - 519.53 + 269.09 = 1022.55\text{K}$$

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For process 3-4 : Isentropic expansion in turbine

$$\frac{T_3}{T_4}=\left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} \quad \therefore p_4=p_3\left(\frac{T_4}{T_3}\right)^{\frac{\gamma}{\gamma-1}}=p_2\left(\frac{T_4}{T_3}\right)^{\frac{\gamma}{\gamma-1}}$$

But $p_2=10\times p_1=10\times 0.492=4.92$ bar

$$\therefore p_4=4.92\times\left(\frac{1022.55}{1273}\right)^{\frac{1.4}{1.4-1}}=2.28 \text{ bar}$$

For process 4-5 : Isentropic expansion in nozzle

$$\frac{T_5}{T_4}=\left(\frac{p_5}{p_4}\right)^{\frac{\gamma-1}{\gamma}} \qquad \therefore T_5=1022.55\times\left(\frac{0.32}{2.28}\right)^{\frac{1.4-1}{1.4}}=583.48 \text{ K}$$

(i) Velocity of gases at nozzle exit

$$C_j=\sqrt{2\eta_n C_p(T_4-T_5)}$$

$$=\sqrt{2\times 1\times 1005\times(1022.55-583.48)}\text{ , (Assuming } \eta_n=1)$$

$$\mathbf{= 939.42 \text{ m/s}}$$ ***Ans***

(ii) Propulsive efficiency

$$\eta_p=\frac{2C_a}{C_j+C_a} \quad \text{(neglecting mass of fuel)}$$

$$=\frac{2\times 250}{939.42+250}\times 100=\mathbf{42.03\%}$$ ***Ans***

(iii) Thermal efficiency

$$\eta_{th}=\frac{\text{propulsive power}}{\text{heat supplied}}$$

$$=\frac{\left(C_j^2-C_a^2\right)}{C_p(T_3-T_2)}=\frac{(939.42)^2-(250)^2}{2\times 1005\times(1273-519.53)}=\mathbf{54.14\ \%}$$ ***Ans***

Problem 11.3 *A turbojet engine flying at a speed of 1000 km/hr, takes air at a rate of 60 kg/s. The air fuel ratio is 70:1. The calorific value of fuel 46,000 kJ/kg. The temperature difference of gases entering and leaving the nozzle is 180°C, and the nozzle efficiency is 90%. Calculate exit velocity of jet, thermal efficiency, propulsive efficiency, overall efficiency of plant and thrust specific fuel consumption. Take for gases $C_p = 1.05$ kJ/kg K.*

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Solution : Given data : $C_a = 1000 \text{ km/hr} = \dfrac{1000\times1000}{3600} = 277.8 \text{ m/s}$, $m_a = 60 \text{ kg/s}$,

A/F = 70 : 1, CV = 46,000 kJ/kg, $T_4' - T_5 = 180^\circ C$, $\eta_n = 0.9$, $C_j = ?$, $\eta_{th} = ?$, $\eta_p = ?$,

$\eta_{overall} = ?$

Exit velocity of jet $C_j = \sqrt{2\eta_n C_p (T_4' - T_5)}$

$$= \sqrt{2\times0.9\times1.05\times10^3 (180)} = \mathbf{583.26 \text{ m/s}} \quad \textit{Ans}$$

Propulsive power $PP = \left(1+\dfrac{m_f}{m_a}\right)\dfrac{C_j^2}{2} - \dfrac{C_a^2}{2}$, W/kg of air

$$= \left(1+\frac{1}{70}\right)\times\frac{(583.26)^2}{2} - \frac{(277.8)^2}{2} = 133.94 \text{ kW/kg of air}$$

Heat supplied rate $Q_s = m_f \cdot CV$, W

$$= \frac{m_f}{m_a}\cdot CV, \text{ W/kg of air}$$

$$= \frac{1}{70}\times46,000 = 657.71 \text{ kW/kg of air}$$

Thermal efficiency $\eta_{th} = \dfrac{PP}{Q_s} = \dfrac{133.94}{657.71}\times100 = \mathbf{20.38\%}$ *Ans*

Thrust produced $T = \left[\left(1+\dfrac{m_f}{m_a}\right)C_j - C_a\right] = \left[\left(1+\dfrac{1}{70}\right)\times(583.26) - 277.8\right]$

$= 313.79 \text{ N/kg of air/s}$

$= m_a \times 313.79 = 60\times313.79 = 18827.53 \text{ N}$

Thrust $T.P = \left[\left(1+\dfrac{m_f}{m_a}\right)C_j - C_a\right]\times C_a = T\times C_a = 313.79 \times 277.8$

$= 87.171 \text{ kW/kg of air}$

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Propulsive efficiency $\eta_p = \frac{T.P.}{P.P.} = \frac{87.171}{133.94} \times 100 = \mathbf{65.08\%}$ **Ans**

Overall efficiency $\eta_{overall} = \eta_{th} \times \eta_p = 0.2038 \times 0.6508 = \mathbf{13.26\%}$ **Ans**

Mass flow rate of fuel $m_f = \frac{m_a}{A/F} = \frac{60}{70} = 0.857$ kg/s

Thrust sfc $= \frac{m_f}{T} = \frac{0.857}{18827.53} = \mathbf{4.55 \times 10^{-5}}$ **kg/Ns** **Ans**

Problem 11.4 *A turbojet engine flying at an altitude of 1000 m with velocity of 250 m/s takes air at 0.3 bar pressure and –47°C temperature and develops thrust power of 800 kW. Temperature of gases at inlet of turbine is 700°C. The velocity of gases at inlet of nozzle is 220 m/s. The pressure ratio in the compression process is 5.2. The efficiencies of compressor, turbine and nozzle are 0.85, 0.86, and 0.9 respectively.*
Calculate (i) air fuel ratio, (ii) mass flow rate of air, (iii) overall thermal efficiency, (iv) outlet area of jet and (v) thrust specific fuel consumption.
Take, for air $C_p = 1.005$ *kJ/kg K,* $\gamma = 1.4$, *for gases* $C_p = 1.08$ *kJ/kg K,* $\gamma = 1.33$
Calorific value of fuel is 44,000 kJ/kg

Solution : Given data : H = 1000 m, $C_a = 250$ m/s, $p_1 = 0.3$ bar, $T_1 = -47°C = 226$ K, TP = 80 kW, $T_3 = 700°C = 973$ K, $C_4 = 220$ m/s, $p_2 / p_1 = 5.2$, $\eta_c = 0.85$, $\eta_t = 0.86$, $\eta_n = 0.9$, A/F = ?, m_a = ?, $\eta_{overall}$ = ?, A_5 = ?, Thrust sfc = ?

For process 1-2 : Isentropic compression (Refer Fig. 11.12)

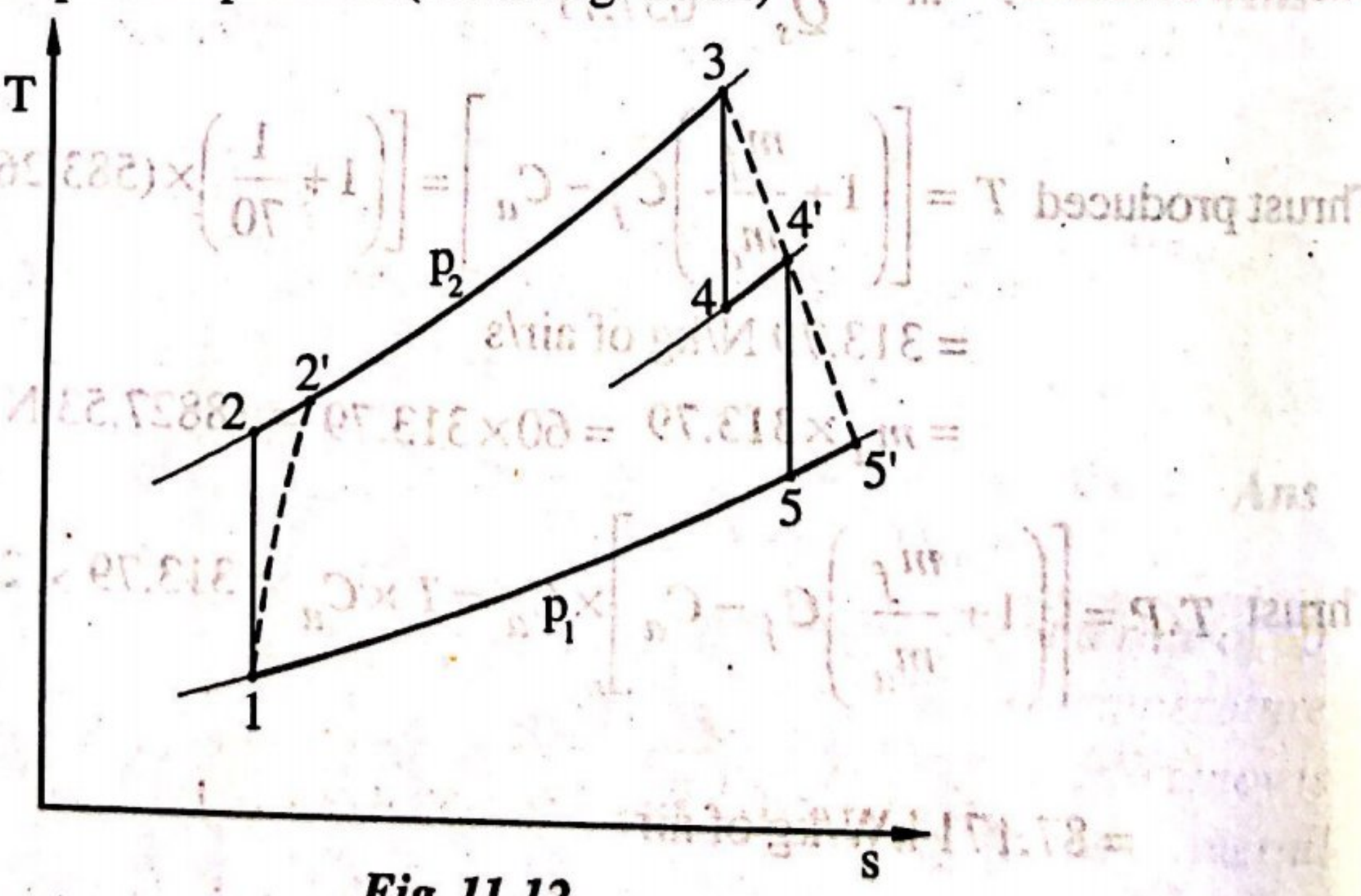


Fig 11.12

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$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 226\times(5.2)^{\frac{1.4-1}{1.4}} = 361.97\,K$$

Now, $\eta_c = \dfrac{T_2 - T_1}{T_2' - T_1}$ $\quad \therefore T_2' = 226 + \dfrac{361.97 - 226}{0.85} = 385.97\,K$

Compressor work input $w_c = C_p(T_2' - T_1) = 1.005\times(385.97 - 226) = 160.77$ kJ/kg of air

For turbojet engine $w_c = w_T$, but $w_T = C_p(T_3 - T_4')$

$\therefore 160.77 = 1.08(973 - T_4')$ $\quad \therefore T_4' = 824.13\,K$

Now, $\eta_t = \dfrac{T_3 - T_4'}{T_3 - T_4}$ $\quad \therefore T_4 = 973 - \dfrac{973 - 824.13}{0.86} = 799.89\,K$

For process 3-4 : Isentropic expansion in turbine

$$\therefore \frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{p_2}{p_4}\right)^{\frac{\gamma-1}{\gamma}} \quad \therefore \frac{p_2}{p_4} = \left(\frac{973}{799.89}\right)^{\frac{1.33}{1.33-1}} = 2.2$$

$$\therefore p_4 = p_2\times 0.454 = \frac{p_2}{p_1}\times p_1 \times 0.454 = 5.2\times 0.3\times 0.454 = 0.708 \text{ bar}$$

For process 4-5 : Isentropic expansion in nozzle

$$\therefore \frac{T_4}{T_5} = \left(\frac{p_4}{p_5}\right)^{\frac{\gamma-1}{\gamma}} \quad \therefore T_5 = T_4\left(\frac{p_5}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = 799.89\times\left(\frac{0.3}{0.708}\right)^{\frac{1.33-1}{1.33}} = 646.4\,K$$

$$\text{But } \eta_n = 0.9 = \frac{\text{Final K.E. of jet}}{\text{Isentropic enthalpy drop in nozzle + K.E. of gases at inlet of nozzle}}$$

$$\therefore 0.9 = \frac{C_j^2/2}{C_p(T_4' - T_5) + C_4^2/2}$$

$$\therefore 0.9 = \frac{C_j^2}{2\times 1.08\times 10^3 (824.13 - 646.4) + (220)^2}$$

$$\therefore C_j = 623.75 \text{ m/s}$$

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Now, actual enthalpy drop in nozzle = change in KE in nozzle

$$\therefore C_p(T_4' - T_5') = \frac{C_j^2 - C_4^2}{2}$$

$$\therefore 1.08\times10^3(824.13 - T_5') = \frac{(623.75)^2 - (220)^2}{2}$$

$$\therefore T_5' = 666.41\text{K}$$

For process 2-3 : Combustion process

Energy balance, $m_f \times CV = (m_a + m_f)C_p(T_3 - T_2')$

$$\therefore CV = \left(\frac{m_a}{m_f} + 1\right)C_p(T_3 - T_2')$$

$$\therefore 44{,}000 = \left(\frac{m_a}{m_f} + 1\right)\times1.08\times(973 - 385.97)$$

$$\therefore \frac{m_a}{m_f} = 68.4$$

(i) Air-fuel ratio $m_a / m_f = \mathbf{68.4}$ ***Ans***

(ii) Mass flow rate of air

Thrust power $T.P. = [(m_a + m_f)C_j - m_a C_a]\times C_a$

$$\therefore 800\times10^3 = m_a \times\left[\left(1 + \frac{m_f}{m_a}\right)C_j - C_a\right]\times C_a$$

$$\therefore 800\times10^3 = m_a \times\left[\left(1 + \frac{1}{68.4}\right)\times623.75 - 250\right]\times250$$

$$\therefore m_a = \mathbf{8.36\ kg/s}$$ ***Ans***

(iii) Overall thermal efficiency

$$\eta_{overall} = \eta_{th}\times\eta_p = \frac{PP}{Q_s}\times\frac{T.P.}{PP} = \frac{TP}{Q_s} = \frac{T.P}{m_f \cdot CV}$$

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$$\therefore \eta_{overall} = \frac{800}{m_a\left(\frac{m_f}{m_a}\times CV\right)} = \frac{800}{8.36\left(\frac{1}{68.4}\times 44{,}000\right)}\times 100 = \mathbf{14.87\ \%}$$

(iv) Outlet area of jet

$$\text{Density of exhaust gases } \rho_g = \frac{p_5}{RT_5'} = \frac{0.3\times 10^5}{0.287\times 10^3\times 666.41} = 0.1568\ \text{kg/m}^3$$

$$\text{Mass flow rate of gases} = \rho_g \times A_5 \times C_j$$

$$\therefore (m_a + m_f) = \rho_g \times A_5 \times C_j$$

$$\therefore m_a\left(1+\frac{m_f}{m_a}\right) = \rho_g \times A_5 \times C_j$$

$$\therefore 8.36\times\left(1+\frac{1}{68.4}\right) = 0.1568\times A_5 \times 623.75$$

$$\therefore A_5 = \text{area of jet at outlet} = \mathbf{0.086\ m^2} \quad \textbf{Ans}$$

(v) Thrust specific fuel consumption

$$T.P. = T\times C_a \quad \therefore \text{Thrust } T = \frac{800\times 10^3}{250} = 3200\ \text{N}$$

$$m_f = m_a \times \frac{m_f}{m_a} = 8.36\times\frac{1}{68.4} = 0.122\ \text{kg/s}$$

$$\text{Thrust sfc} = \frac{m_f}{T} = \frac{0.122}{3200} = \mathbf{3.82\times 10^{-5}\ kg/Ns} \quad \textbf{Ans}$$

Problem 11.5 *A jet propulsion unit takes air at a pressure 0.5 bar and temperature 15°C. Then the air is compressed to pressure 3 bar in the compressor. The final temperature is 1.2 times that for isentropic compression. The temperature of combustion product at inlet of turbine is 600°C. Assume that turbine work produced is equal to compressor work input, the gases expanded isentropically in both turbine and nozzle. Neglect the velocity of approach. Calculate (i) Power required to drive compressor, (ii) Air-fuel ratio if the calorific value of fuel is 44,000 kJ/kg, and (iii) Static thrust produced per kg of air per second.*

Solution : Given data : $p_1 = 0.5$ bar, $T_1 = 15^oC = 288K$, $p_2 = 3$ bar,

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$T_2' - T_1 = 1.2(T_2 - T_1)$, $T_3 = 600^o C = 873\,K$, Velocity of approach $= C_a \cong 0$, $w_c = ?$, A/F = ?, T = ?

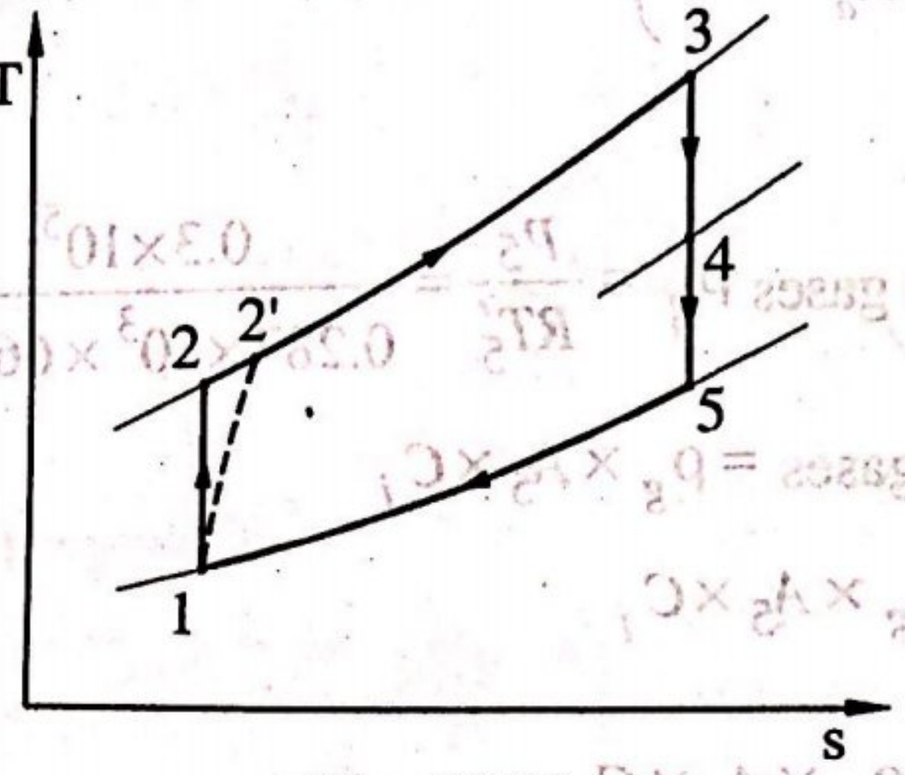


Fig. 11.13

For process 1-2 : Isentropic compression (Refer Fig. 11.13)

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 288\left(\frac{3}{0.5}\right)^{\frac{1.4-1}{1.4}} = 480.53\,K$$

But, $T_2' - T_1 = 1.2(T_2 - T_1) = 1.2(480.53 - 288) = 231.03$

$$\therefore\ T_2' = 519.03\,K$$

Work consumed by compressor

$$w_c = C_p(T_2' - T_1) = 1.005(519.03 - 288) = \mathbf{232.19\ kJ/kg\ of\ air}$$ ***Ans***

Now, heat supplied in combustion chamber

$$q_s = C_p(T_3 - T_2') = 1.005(873 - 519.03) = 355.74\ \text{kJ/kg of air}$$

But $q_s = m_f \times CV$, kJ

$$\therefore 355.74 \times m_a = m_f \times CV$$

$$\therefore \frac{m_a}{m_f} = \frac{44,000}{355.74} = 123.68$$

$\therefore$ Air-fuel ratio = **123.68 : 1** ***Ans***

Since, $w_c = w_T$

$$\therefore C_p(T_2' - T_1) = C_p(T_3 - T_4)$$

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$\therefore 519.03 - 288 = 873 - T_4 \quad \therefore T_4 = 641.97 \text{ K}$

For isentropic expansion process 3-5 :

$$\frac{T_3}{T_5} = \left(\frac{p_3}{p_5}\right)^{\frac{\gamma-1}{\gamma}} \quad \therefore T_5 = T_3 \times \left(\frac{p_5}{p_3}\right)^{\frac{\gamma-1}{\gamma}} = 873 \times \left(\frac{0.5}{3}\right)^{\frac{1.4-1}{1.4}} = 523.22\,K$$

Now, velocity of jet $C_j = \sqrt{2\eta_n C_p (T_4 - T_5)}$

$$= \sqrt{2 \times 1 \times 1.005 \times 10^3 (641.97 - 523.22)} = 488.55 \text{ m/s}$$

Thrust produced $T = \left(1 + \frac{m_f}{m_a}\right) C_j - C_a$, N/kg of air/s

$$= \left(1 + \frac{1}{123.68}\right) \times 488.55 - 0 = \textbf{492.5 N/kg of air/s} \quad \textit{Ans}$$

Problem 11.6 *A turbojet engine travels at 200 m/s in air at pressure 0.7 bar and temperature –7°C. Air first enters diffuser in which it is brought to rest relative to the unit and it is then compressed in a compressor through a pressure ratio 6 and fed to a turbine at 1000°C. The gases expand through the turbine and then through the nozzle to atmospheric pressure 0.7 bar. The efficiencies of diffuser, nozzle, and compressor are each 90%. The efficiency of turbine is 82%. The pressure drop in the combustion chamber is 0.12 bar.*

Calculate (i) Air fuel ratio, (ii) Specific thrust of the unit, and (iii) Total thrust, if the inlet cross section of diffuser is 0.12 m². Assume calorific value of fuel as 44,000 kJ/kg.

Solution : Given data : $C_a = 200$ m/s, $p_0 = 0.7$ bar, $T_0 = -7°C = 266$ K, pressure ratio in compressor = 6, $T_3 = 1000°C = 1273$ K, pressure drop in combustion chamber = 0.12 bar, $p_5 = p_0$, $\eta_d = \eta_c = \eta_n = 0.9$, $\eta_t = 0.82$, A/F = ?, Specific thrust = ?, Total thrust = ?, $A_0 = 0.12$ m², CV = 44,000 kJ/kg.

(i) Air fuel ratio (A/F)

For ideal diffuser, process 0-1 (Refer Fig. 11.14), energy equation is given by

$$h_1 = h_0 + \frac{C_a^2}{2} \quad \therefore T_1 = T_0 + \frac{C_a^2}{2C_p}$$

$$\therefore T_1 = 266 + \frac{200^2}{2 \times 1.005 \times 10^3} = 285.9\,K$$

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Now, diffuser efficiency $\eta_d = 0.9 = \dfrac{h_1 - h_0}{h_1' - h_0}$

$$\therefore h_1' = h_0 + \frac{h_1 - h_0}{0.9} = h_0 + \frac{C_a^2}{2 \times 0.9}$$

$$\therefore T_1' = T_0 + \frac{C_a^2}{2 \times 0.9 \times C_p} = 266 + \frac{(200)^2}{2 \times 0.9 \times 1.005 \times 10^3} = 288.11\,K$$

Also, $\dfrac{T_1}{T_0} = \left(\dfrac{p_1}{p_0}\right)^{\frac{\gamma-1}{\gamma}}$ $\therefore p_1 = p_0 \left(\dfrac{T_1}{T_0}\right)^{\frac{\gamma}{\gamma-1}} = 0.7 \times \left(\dfrac{285.9}{266}\right)^{\frac{1.4}{1.4-1}} = 0.9$ bar

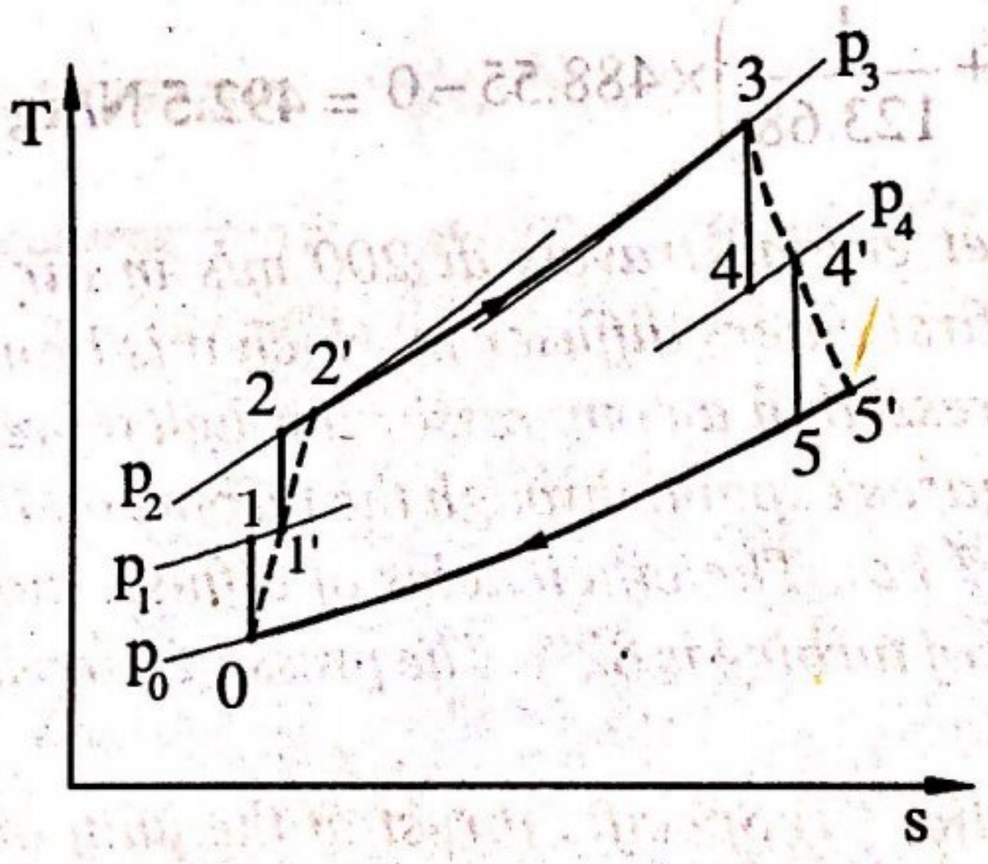


Fig. 11.14

For compressor, $\dfrac{T_2}{T_1'} = \left(\dfrac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$ $\therefore T_2 = 288.11 \times (6)^{\frac{1.4-1}{1.4}} = 480.71\text{K}$

Also $\eta_C = 0.9 = \dfrac{T_2 - T_1'}{T_2' - T_1'}$

$$\therefore T_2' = T_1' + \frac{T_2 - T_1'}{0.9} = 288.11 + \left(\frac{480.71 - 288.11}{0.9}\right) = 502.11\text{K}$$

Heat supplied to combustion chamber

$$Q_s = m_f \times CV = (m_a + m_f) C_p T_3 - m_a C_p T_2'$$

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$$\therefore CV = \left(\frac{m_a}{m_f} + 1\right) C_p T_3 - \frac{m_a}{m_f} C_p T_2'$$

$$\therefore CV = \frac{m_a}{m_f} C_p (T_3 - T_2') + C_p T_3$$

$$\therefore 44{,}000 = \frac{m_a}{m_f} \times 1.005 (1273 - 502.11) + 1.005 \times (1273)$$

$$\therefore \frac{m_a}{m_f} = \text{Air fuel ratio} = \mathbf{55.14} \qquad \textit{Ans}$$

(ii) Specific thrust of the unit

Pressure at inlet of turbine $p_3 = p_2 -$ pressure drop in combustion chamber

$$\doteq (p_1 \times 6) - 0.12$$

$$= (0.9 \times 6) - 0.12 = 5.28 \text{ bar}$$

For turbojet work produced by turbine = work input to compressor

$$\therefore (m_a + m_f) C_p (T_3 - T_4') = m_a C_p (T_2' - T_1')$$

$$\therefore \left(1 + \frac{m_f}{m_a}\right)(T_3 - T_4') = (T_2' - T_1')$$

$$\therefore \left(1 + \frac{1}{55.14}\right)(T_3 - T_4') = 502.11 - 288.11$$

$$\therefore T_3 - T_4' = 210.18 \qquad \therefore T_4' = 1273 - 210.18 = 1062.81 \text{K}$$

Now, for turbine $\eta_t = 0.82 = \dfrac{T_3 - T_4'}{T_3 - T_4}$

$$\therefore T_4 = T_3 - \frac{(T_3 - T_4')}{0.82} = 1273 - \frac{(1273 - 1062.81)}{0.82} = 1016.67 \text{K}$$

Also, $\dfrac{T_3}{T_4} = \left(\dfrac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}}$ $\therefore p_4 = p_3 \times \left(\dfrac{T_4}{T_3}\right)^{\frac{\gamma}{\gamma-1}} = 5.28 \times \left(\dfrac{1016.67}{1273}\right)^{\frac{1.4}{1.4-1}} = 2.4$ bar

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For nozzle, $\frac{T_4'}{T_5} = \left(\frac{p_4}{p_5}\right)^{\frac{\gamma-1}{\gamma}}$ $\therefore T_5 = T_4' \times \left(\frac{p_5}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = 1062.81 \times \left(\frac{0.7}{2.4}\right)^{\frac{1.4-1}{1.4}} = 747.42\,\text{K}$

Also, $\eta_n = 0.9 = \frac{T_4' - T_5'}{T_4' - T_5}$

$$\therefore T_5' = T_4' - 0.9(T_4' - T_5) = 1062.81 - 0.9(1062.81 - 747.42) = 778.96\ \text{K}$$

Velocity of gases at the exit of the nozzle

$$C_j = \sqrt{2\eta_n C_p (T_4' - T_5)}$$

$$= \sqrt{2C_p (T_4' - T_5')} = \sqrt{2 \times 1.005 \times 10^3 (1062.81 - 778.96)}$$

$$= 755.33\ \text{m/s}$$

Specific thrust $T = \left(1 + \frac{m_f}{m_a}\right) C_j = C_a$, N/kg of air/s

$$= \left(1 + \frac{1}{55.14}\right) \times 755.33 - 200 = \textbf{569.036 N/kg of air/s}$$ ***Ans***

(iii) Total thrust

Volume of air entering the turbojet per second

V_0 = Inlet area of diffuser × Air velocity

$= 0.12 \times 200 = 24\ \text{m}^3/\text{s}$

Now, mass of air entering the turbojet per second

$$m_a = \frac{p_0 V_0}{R T_0} = \frac{0.7 \times 10^5 \times 24}{287 \times 266} = 22\,kg/s$$

$\therefore$ Total thrust = m_a × specific thrust

$= 22 \times 569.036 = \textbf{12522.34 N}$ ***Ans***

Problem 11.7 *A turbojet engine inducts 45 kg of air per second and propels an aircraft with a uniform flight speed of 880 km/h. The isentropic enthalpy change for nozzle is 188.37 kJ/kg and its velocity co-efficient is 0.96. The fuel air ratio is 0.012, the combustion efficiency is 0.95 and the lower heating value of fuel is 44,000 kJ/kg.*
Calculate (a) the thermal efficiency of engine (b) fuel flow rate in kg/h and TSFC, (c) the propulsion power in kW, (d) the thrust power, (e) the propulsive efficiency and (f) the overall efficiency. ***[same as May '14, Nov. '13]***

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Solution : Given data : m_a = 45 kg/s, C_a = 880 km/hr = 244.44 *m/s*, $h_4' - h_5 = 188.37\ kJ/kg$, $K_n = 0.96$, $\eta_{comb.} = 0.95$, *F/A* = 0.012, C.V. = 44,000 kJ/kg, η_{th} = ?, m_f = ?, TSFC = ?, PP = ?, TP = ?, η_p = ?, $\eta_{overall}$ = ?.

The velocity of gases leaving the nozzle

$$C_j = \sqrt{2\eta_n C_p (T_4' - T_5)}$$

$$= \sqrt{2\eta_n (h_4' - h_5)} = \sqrt{2K_n^2 (h_4' - h_5)} \quad (\because \eta_n = K_n^2, \text{ where } K_n = \text{velocity co-efficient})$$

$$= \sqrt{2 \times (9.96)^2 (188.37 \times 10^3)}$$

$$= 589.24 \text{ m/s}$$

(a) Thermal efficiency of the engine

$$\eta_{th} = \frac{\left(1 + \frac{m_f}{m_a}\right)\frac{C_j^2}{2} - \frac{C_a^2}{2}}{\frac{m_f}{m_a} \times CV} = \frac{(1+0.012) \times \left(\frac{(589.24)^2}{2} - \frac{(244.44)^2}{2}\right)}{0.012 \times 44000 \times 10^3}$$

$$= \mathbf{27.61\%} \qquad \textit{Ans}$$

(b) Mass of fuel supplied $m_f = \left(\frac{m_f}{m_a}\right) \times m_a$

$$= 0.012 \times 45 = 0.54\ kg/s$$

$$= \mathbf{1944\ kg/h} \qquad \textit{Ans}$$

Thrust specific Fuel consumption (TSFC)

$$TSFC = \frac{m_f}{T} = \frac{m_f}{m_a\left[\left(1 + \frac{m_f}{m_a}\right)C_j - C_a\right]}\ kg/Ns$$

$$= \frac{0.012}{(1+0.012) \times 589.24 - 244.44}$$

$$= \mathbf{3.4 \times 10^{-5}\ kg/Ns} \qquad \textit{Ans}$$

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(c) Propulsive power (PP)

$$PP = m_a\left[\left(1+\frac{m_f}{m_a}\right)\frac{C_j^2}{2}-\frac{C_a^2}{2}\right] Watt$$

$$= 45\times\left[(1+0.012)\frac{(589.24)^2}{2}-\frac{(244.44)^2}{2}\right]$$

$$= \mathbf{5217.04\ kW} \quad \textit{Ans}$$

(d) Thrust power $(TP) = m_a\left[\left(1+\frac{m_f}{m_a}\right)C_j - C_a\right]\times C_a$

$$= 45\times[(1+0.012)589.24-244.44]\times 244.44$$

$$= \mathbf{3870.51\ kW} \quad \textit{Ans}$$

(e) Propulsive efficiency $\eta_p = \frac{TP}{PP} = \frac{3870.51}{5217.04}\times 100$

$$= \mathbf{74.19\%} \quad \textit{Ans}$$

- (f) Overall efficiency $\eta_{overall} = \eta_{th}\times\eta_p = 0.2761\times 0.7419$

$$= \mathbf{20.48\ \%} \quad \textit{Ans}$$

11.11 Rocket engine

If the propulsion unit contains its own oxygen supply for combustion purposes, the system is known as ***rocket engine*** or ***rocket propulsion***. A rocket engine works on same principle as that of jet engine. i.e. the gases expands through the nozzles, and are accelerated to extremely high speed to exert a large reactive thrust on the rocket body (since every action has an equal and opposite reaction). In case of jet engine the oxygen required for combustion is taken from the atmosphere and fuel is stored whereas the rocket engine is non-breathing engine, the fuel and oxidizer both are contained in a propelling body. Hence, rocket engine does not depend upon atmospheric air for the supply of the oxidant. Therefore, rocket engine can be operate at very high altitudes and even in vacuum. The rocket engine is not suitable for low speed use. As compared to other propulsion system, it is very light in weight and powerful.

11.12 Basic theory of operation of rocket engines [Nov. '14]

The rocket engine in its simplest form as shown in Fig. 11.14 consists of rocket body with fuel tank and oxidizer tank, combustion chamber, propellant feeding system and an

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expanding nozzle. The fuel and oxidizer are commonly known as propellants. In the liquid propellants type rocket engine, fuel tank contains alcohol and oxidizer tank contains liquid oxygen.

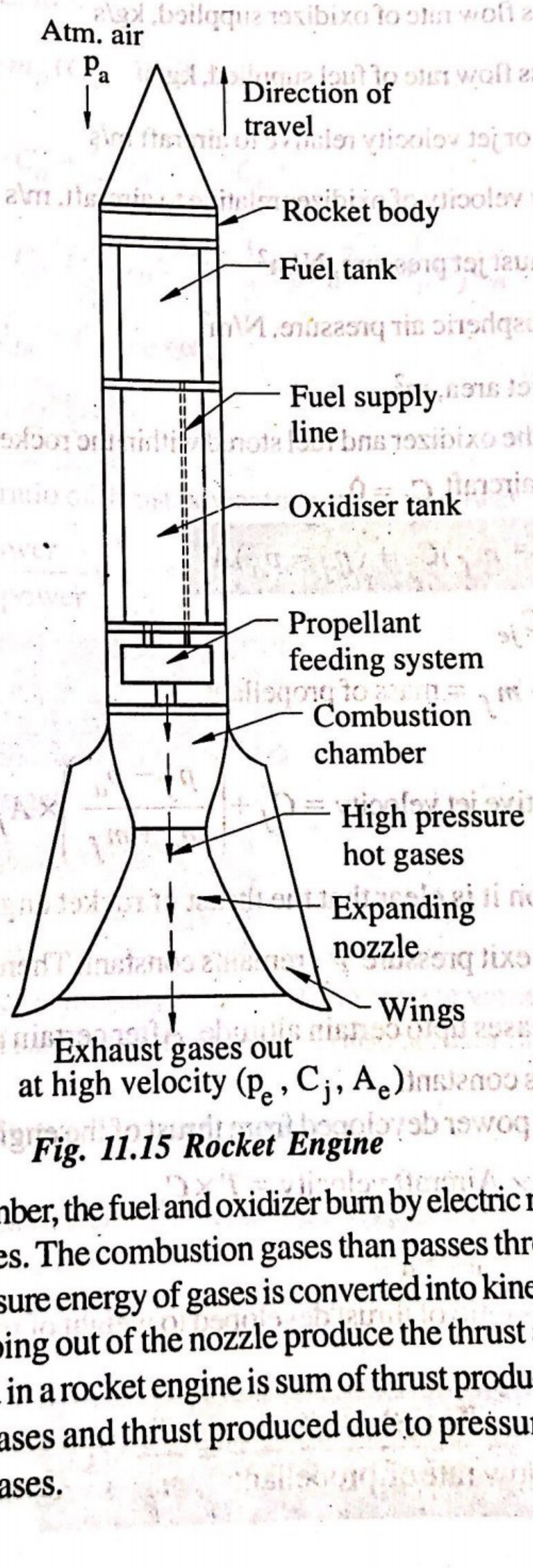


Fig. 11.15 Rocket Engine

In the combustion chamber, the fuel and oxidizer burn by electric means which produces high pressure combustion gases. The combustion gases than passes through the convergent - divergent nozzle in which pressure energy of gases is converted into kinetic energy. These high velocity (supersonic) gases going out of the nozzle produce the thrust and propel the rocket.

Thrust : The thrust developed in a rocket engine is sum of thrust produced due to momentum (kinetic energy) of exhaust gases and thrust produced due to pressure difference between atmospheric air and exhaust gases.

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Thrust = [Momentum thrust] + [Pressure thrust]

$$= [(m_o + m_f) \times (C_j - C_i)] + [(p_j - p_a) \times A_j]$$

where, m_o = mass flow rate of oxidizer supplied, kg/s

m_f = mass flow rate of fuel supplied, kg/s

C_j = exit or jet velocity relative to aircraft m/s

C_i = entry velocity of oxidizer relative to aircraft, m/s

p_j = exhaust jet pressure, N/m^2

p_a = atmospheric air pressure, N/m^2

A_j = exit jet area, m^2

For rocket engine, the oxidizer and fuel stored within the rocket itself, hence entry velocity of oxidizer relative to aircraft $C_i = 0$.

$$\therefore T = (m_o + m_f)C_j + (p_j - p_a)A_j$$

$$\therefore T = m_p C_{je}$$

where, $m_p = m_o + m_f$ = mass of propellant

$$C_{je} = \text{effective jet velocity} = C_j + \left(\frac{p_j - p_a}{m_o + m_f}\right) \times A_j$$

From above equation it is clear that the thrust of rocket engine is independent of aircraft velocity. The nozzle exit pressure p_j remains constant. Therefore, the thrust increases with altitude as p_a decreases upto certain altitude. After certain altitude p_a remains constant, hence thrust becomes constant.

Thrust Power : The power developed from thrust of the engine is called thrust power

$$\text{T.P.} = \text{Thrust} \times \text{Aircraft velocity} = T \times C_a$$

$$\therefore \text{T.P.} = m_p \cdot C_{je} \cdot C_a \quad \ldots(11.28)$$

Specific impulse : It is ratio of thrust developed to weight of propellant which passes through the rocket engine.

$$I_{sp} = \frac{\text{Thrust}}{\text{Mass flow rate of propellant}} = \frac{T}{m_p} = C_{je} \quad \ldots(11.29)$$

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Propulsive power :

Propulsive power = Thrust power + Power loss in exhaust

= TP + Power lost due to energy carried by exhaust

= TP + K.E. loss

$$= \text{T.P.} + \frac{1}{2} m_p (C_j - C_a)^2$$

$$= m_p C_{je} \cdot C_a + \frac{1}{2} m_p (C_j - C_a)^2$$

$$= m_p C_{je} \cdot C_a + \frac{1}{2} m_p C_j^2 + \frac{1}{2} m_p C_a^2 - m_p C_j C_a$$

Neglecting pressure thrust, $C_{je} = C_j$ we get,

$$\therefore \text{P.P.} = = \frac{1}{2} m_p (C_j^2 + C_a^2) \quad \ldots(11.30)$$

Propulsive efficiency : It is ratio of thrust power to propulsive power

$$\eta_p = \frac{\text{Thrust power}}{\text{Propulsive power}}$$

$$\therefore \eta_p = \frac{m_p C_{je} \cdot C_a}{m_p C_{je} \cdot C_a + \frac{1}{2} m_p (C_j - C_a)^2} \quad \ldots(11.31)$$

Neglecting pressure thrust, we get

$$\eta_p = \frac{m_p C_j \cdot C_a}{\frac{1}{2} m_p (C_j^2 - C_a^2)} = \frac{2 C_j C_a}{C_j^2 + C_a^2} \quad \ldots(11.32)$$

Rockets or rocket engines are commonly used to take the space vehicle outside the earth's atmosphere, to launch artificial satellites in the space. It is also used for fireworks and weaponry enjection seats.

Advantages and disadvantages of rocket engines :

Advantages :

(1) It is self contained, and non-air-breathing engine, hence it does not depend on atmospheric air condition.

(2) The thrust produced by rocket engine is increased with altitude and after certain altitude it remains constant.

(3) Rockets are highly efficient at very high speed (about Mach No. > 5)

(4) Rocket engine develops high thrust per unit area. Thrust to weight ratio is over 100.

(5) Rockets can be operate in vacuum.

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Disadvantages :

(1) It requires large amount of propellant (fuel and oxidizer) and has very low specific impulse.

(2) The oxidizer and fuel tank are within the rocket body, makes rocket a very risky vehicle.

(3) Rockets are extremely noisy space vehicle.

(4) It offers extreme thermal stresses of combustion chamber.

Problem 11.8 *A 250 km range flying bomb is travelled at altitude 600 m with an average speed of 180 m/s. It is required thrust of 2700 N. The maximum temperature rise in the combustion chamber is 850°C. The fuel consumed is 450 kg of petrol (CV = 43000 kJ/kg). The diameter of the discharge nozzle is 0.3m. Calculate (i) Air fuel ratio, (ii) Absolute and relative velocity of exhaust gases to the bomb, (iii) Temperature of exhaust gases and (iv) Propulsive and overall efficiency of unit.*

Take C_p *= 1.05 kJ/kg K for gases, barometric pressure of 600 m altitude is 0.9 bar.*

Solution : Given data : Distance travelled = 250 km, $C_a = 180$ m/s, T = 2700 N, Temperature rise in CC = 850°C, $m_f = 450$ kg during 250 km journey, $C_v = 43{,}000$ kJ/kg, A/F = ?, $C_j = ?$, $C_j - C_a = ?$, $T_j = ?$, $\eta_p = ?$, $\eta_o = ?$, $C_p = 1.05$ kJ/kg K

For combustion chamber

Heat supplied = heat gain by air

$$\therefore m_f \times CV = (m_a + m_f) C_p \Delta T$$

$$\therefore CV = \left(\frac{m_a}{m_f} + 1\right) C_p \Delta T$$

$$\therefore 43{,}000 = \left(\frac{m_a}{m_f} + 1\right) \times 1.05 \times 850$$

$$\therefore \frac{m_a}{m_f} = \text{Air-fuel ratio} = \mathbf{47.18 : 1}$$ ***Ans***

$$\text{Now duration of flight} = \frac{\text{Distance travelled}}{\text{Average speed}}$$

$$= \frac{250 \times 10^3}{180} = 1388.89 \text{ second}$$

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Fuel consumption per second $m_f = \frac{450}{1388.89} = 0.324$ kg/s

Air flow rate $m_a = \left(\frac{m_a}{m_f}\right) \times 0.324 = 15.28$ kg/s

Now, Thrust $T = (m_a + m_f)C_j - m_a C_a$

$\therefore 2700 = (15.28 + 0.324)C_j - 15.28 \times 180$

$\therefore C_j =$ **348.15 m/s (relative velocity)** ***Ans***

$\therefore C_j - C_a = 349.15 - 180 =$ **169.15 m/s (absolute velocity)** ***Ans***

The equation of state for exhaust gases

$$m_g = \frac{pV}{RT}$$

$\therefore T_j =$ temperature of exhaust gases $= \frac{pV}{Rm_g}$

But $V = \frac{\pi}{4} d^2 \times C_j = \frac{\pi}{4}(0.3)^2 \times 349.15 = 24.68 m^3/s$

$$\therefore T_j = \frac{0.9 \times 10^5 \times 24.68}{287 \times (15.28 + 0.324)} = \mathbf{495.98\ K}$$ ***Ans***

Now, propulsive efficiency

$$\eta_p = \frac{\left[\left(1 + \frac{m_f}{m_a}\right)C_j - C_a\right] \times C_a}{\left[\left(1 + \frac{m_f}{m_a}\right)\frac{C_j^2}{2} - \frac{C_a^2}{2}\right]}$$

$$= \frac{2 \times \left[\left(1 + \frac{1}{47.18}\right) \times 349.15 - 180\right] \times 180}{\left[\left(1 + \frac{1}{47.18}\right) \times (349.15)^2 - (180)^2\right]} = \mathbf{51.13\%}$$ ***Ans***

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Overall efficiency of unit

$$\eta_o = \frac{\left[\left(1+\frac{m_f}{m_a}\right)C_j - C_a\right]\times C_a}{\frac{m_f}{m_a}\times CV}$$

$$= \frac{\left[\left(1+\frac{1}{47.18}\right)\times(349.15)-180\right]\times 180}{\left(\frac{1}{47.18}\right)\times 43,000\times 10^3} = \mathbf{3.48\ \%} \quad \textit{Ans}$$

Exercises

11.1 What is the principle of jet propulsion ? Explain turboprop engine.
11.2 State advantages and disadvantages of jet propulsion.
11.3 Write short note on Ram jet engine.
11.4 Write short note on Turbojet engine.
11.5 Draw rough sketch showing pressure, temperature and velocity distribution in turbojet engine. Explain turbojet engine with after burner.
11.6 Explain ramjet with T-s diagram.
11.7 Show that the propulsive efficiency of turbojet is given by $\eta_p = \frac{2}{1+C_j/C_a}$ where, C_j = jet velocity, C_a = aircraft velocity. Why propulsive efficiency in the neighborhood of 100% is not used ?
11.8 With the aid of neat diagram explain the working of ramjet engine. What are it merits, demerits and applications ?
11.9 Explain working difference between propeller jet, turbojet and turboprop.
11.10 Compare and differentiate the jet propulsion and rocket propulsion engine.
11.11 Explain basic working principle of rocket engine.
11.12 State the difference between air breathing and non-air breathing engines.

Unsolved Numerical Problems

11.1 An air plane is propelled by turbojet engine. The flight speed at 1200 m altitude is 220 m/s. The turbojet takes air at 0.25 bar pressure and –50°C temperature. The pressure of air after compression is 3.5 bar and the temperature at inlet of turbine is 1000°C. The propulsive efficiency is 50%. Assume isentropic compression in compressor

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and diffuser. Also neglect the mass of fuel supplied. Calculate (i) Exit velocity of jet and (ii) Thermal efficiency of the system. Take $C_p = 1.005$ kJ/kg K, $\gamma = 1.4$ for air.

[**Ans :** (i) 660 m/s, (ii) 24.2%]

11.2 A turbojet engine propelled air plane flying at an altitude of 10,000 m with speed 800 km/hr. The density of air at 10,000 m altitude is 0.16 kg/m^3. The drag on plane and propulsive efficiency are 6200 N and 52% respectively. The overall efficiency of the turbine plant is 21%. The calorific value of fuel is 45,500 kJ/kg. Calculate (i) absolute velocity of the jet, (ii) volume of air compressed per minute, (iii) diameter of jet, (iv) air fuel ratio, (v) power output of unit.

[**Ans** : (i) 632.27 m/s, (ii) 5667.71 m^3/min, (iii) 0.436 m, (iv) 54.5 : 1, (v) 2649.57 kW]

11.3 Air enters a turbojet engine at 0.8 bar, –32°C and an inlet velocity of 275 m/s. The pressure ratio across the compressor is 8. The turbine inlet temperature is 920°C and the pressure at the nozzle exit is 0.8 bar. The work developed by the turbine equals the compressor work input. The diffuser, compressor, turbine and nozzle processes are isentropic. Determine the nozzle exit velocity of gases. Assume no pressure drop in combustion chamber.

[**Ans** : 893.33 m/s]

11.4 In a jet propulsion unit air is drawn into the rotary compressor at 12°C and 1 bar and delivered at 4 bar. The isentropic efficiency of compression is 82%. After delivery the air is heated at constant pressure until the temperature reaches 800°C. The air then passes through a turbine unit which drives the compressor only and has an isentropic efficiency of 80% before passing through the nozzle and expanding to atmospheric pressure of 1 bar with an efficiency 90%. Neglect mass increase due to the weight of fuel and effect of the velocity of approach.

Calculate : (i) the power required to drive the compressor. (ii) the air fuel ratio if the fuel has a calorific value of 42500 kJ/kg. (iii) the thrust per kg of air per second.

Take $\gamma = 1.4$ and $R = 0.287$ kJ/kg K for air and gases.

[**Ans :** (i) 169.67 kW/kg of air per second, (ii) 67.34 : 1, (iii) 514.9 N/kg of air per sec.]

11.5 A turbojet engine takes 50 kg/sec of air and propels an aircraft with uniform flight speed of 900 km/hr. Isentropic enthalpy change for nozzle is 182 kJ/kg and its efficiency is 0.92. The air fuel ratio 70:1. The combustion efficiency is 95% and calorific value of fuel is 46,000 kJ/kg. Calculate (i) thermal efficiency of engine, (ii) fuel flow in kg/s and thrust specific fuel consumption, (iii) propulsive power in kW, (iv) thrust power in kW, (v) propulsive and overall efficiencies.

[**Ans :** (i) 20.8%, (ii) 0.71 kg/s, 4.28×10^{-5} kg/N of thrust/sec, (iii) 6835.74 kW, (iv) 4171.5 kW, (v) 61%, 12.7%]

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Objective type questions

1. A jet engine is _____
   (a) air-breathing engine (b) non-air-breathing engine
   (c) carries own oxidizer (d) works in vacuum
2. A jet engine has
   (a) propeller at front (b) propeller at back
   (c) propeller on top (c) no propeller
3. Which of the following is not advantages of jet engine ?
   (a) Specific weight of engine is 1/4th to 1/2th that of I.C. engine.
   (b) Thrust produced by engine does not depend on altitude.
   (c) Thermal efficiency is greater than rocket engine
   (d) none of these
4. The efficiency of a jet engine is higher at
   (a) low speed (b) low altitude
   (c) high speed (d) high altitude
5. Which of the following is not correct with reference to jet propulsion
   (a) Propulsive efficiency $= \frac{\text{Thrust power}}{\text{Propulsive power}}$
   (b) Thermal efficiency $= \frac{\text{Propulsive power}}{\text{Heat supply rate}}$
   (c) Overall efficiency $= \frac{\text{Thrust power}}{\text{Heat supply rate}}$
   (d) none of these above
6. A turbo-prop is preferred to a turbojet engine because
   (a) it can fly at high altitude (b) it can fly at high speed
   (c) it has high power for take-off (d) it has high propulsive efficiency at high speed
7. A jet engine uses oxygen for combustion of its fuel from
   (a) oxidizer (b) surrounding air
   (c) propellent (d) none of these above
8. A rocket engine uses oxygen for combustion of its fuel from
   (a) oxidizer (b) surrounding air
   (c) propellent (d) none of these above
9. Which of following statement is not correct with reference to ramjet ?
   (a) At low speed fuel comsumption is high.
   (b) Thrust developed at high altitude is reduced
   (c) It can be operated under the static condition.

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(d) It does not have any moving parts.

10. In a turbojet engine the diffuser is fitted at
    (a) after compressor (b) before turbine
    (c) after turbine (d) nose

11. A rocket engine can be propelled to space because
    (a) it is not air breathing engine (b) it produces very high thrust
    (c) it has high propulsive efficiency (d) none of the above

Answers

1. (a) **2.** (c) **3.** (b) **4.** (b) **5.** (d) **6.** (c) **7.** (b) **8.** (a) **9.** (c) **10.** (d) **11.** (a)

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Chapter 12

Economics of Power generation

12.1 Introduction
12.2 Load curves
12.3 Load duration curves
12.4 Terms and definitions
12.5 Importance of Load factor and diversity factor
12.6 Base load and peak load power plants
12.7 Economic load sharing between base load and peak load power stations
12.8 Cost of power plant
12.9 Performance and operating characteristics of power plant
12.10 Tariff for electric energy

12.1 Introduction

In all fields of industrial production, economics plays an important role. In power plant engineering, economics of power system use certain well established techniques for choosing the most suitable system. The power plant design must be made on the basis of most economical condition and not on the most efficient condition as the profit is the main basis in the design of the plant and its effectiveness is measured financially. The main purpose of design and operation of the plant is to bring the cost of energy produced to minimum. Among many factors, the efficiency of the plant is one of the factors that determines the energy cost. In majority of cases, unfortunately, the most thermally efficient plant is not economic one.

12.2 Load curves

The load on power plants will always be changing with time and will not be constant because consumer of electric power will use the power as and when required.

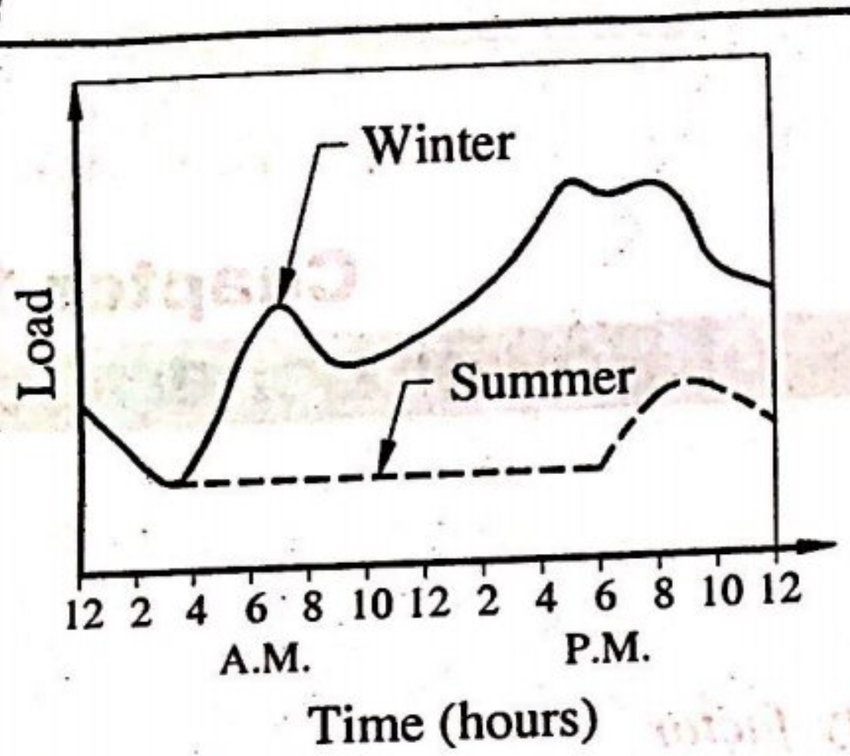
Load curve is graphical representation between load in kW and time. It shows variation of load on the power station. *If the time is in hours then the load curve is known as* ***daily load curve***. *If the time is in days, the load curve is known as* ***monthly load curve*** and *if the time is in months, the load curve is known as* ***yearly or annual load curve***.

The daily load curve will be different for different type of consumers and different localities. These load curves may show different pattern during summer, winter and rainy season.

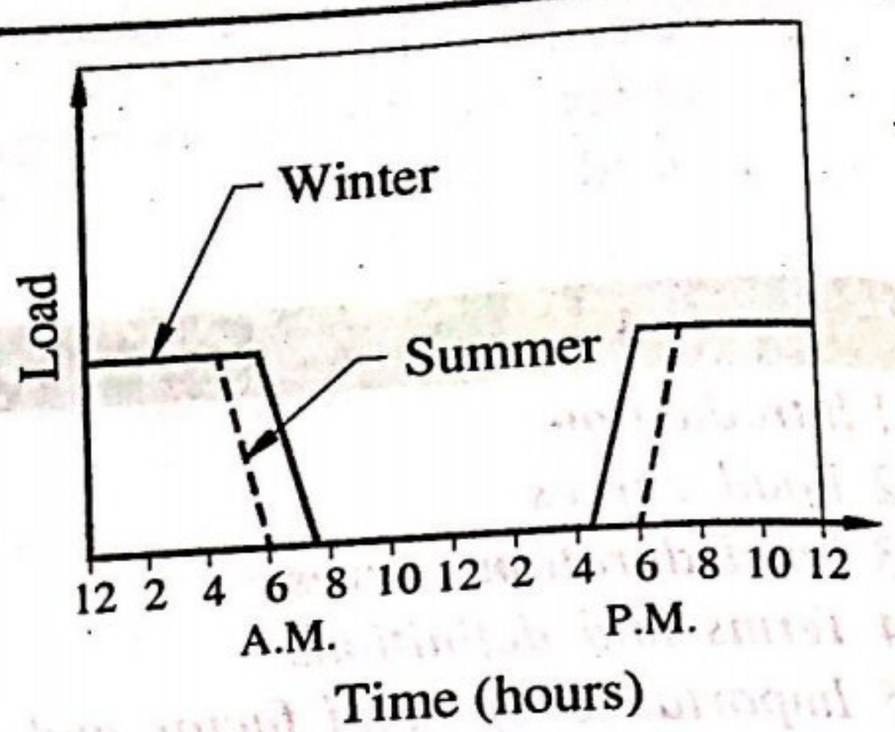
Few daily load curves for industrial, commercial, domestic, street lighting and urban traction are shown in Fig. 12.1.

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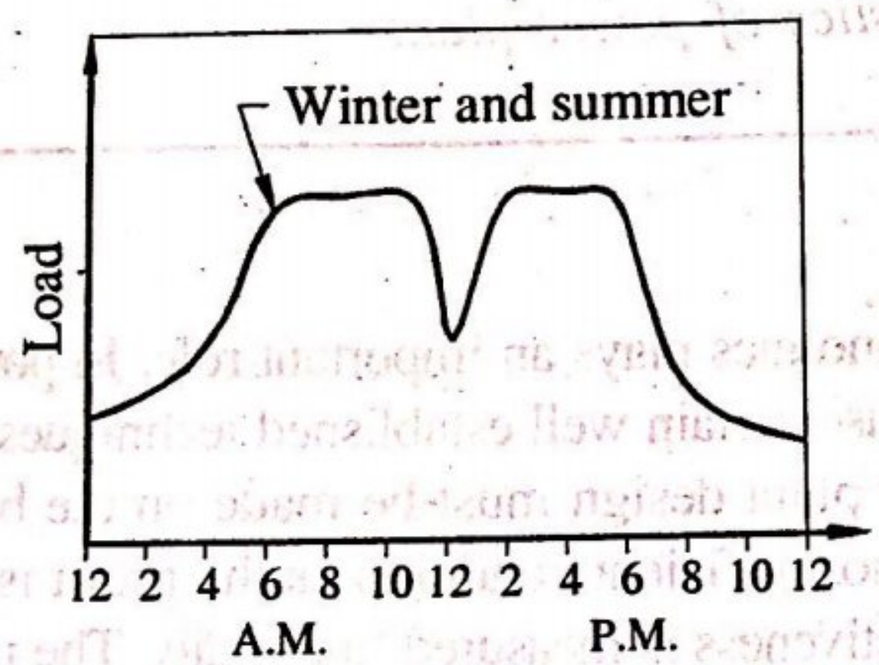
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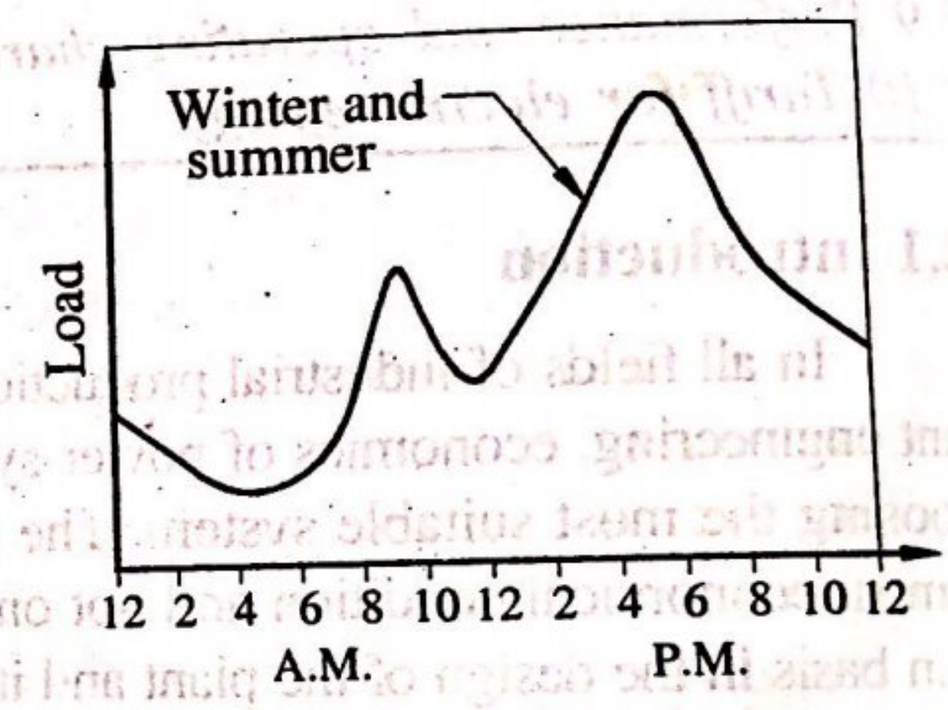
(a) Residential load curve



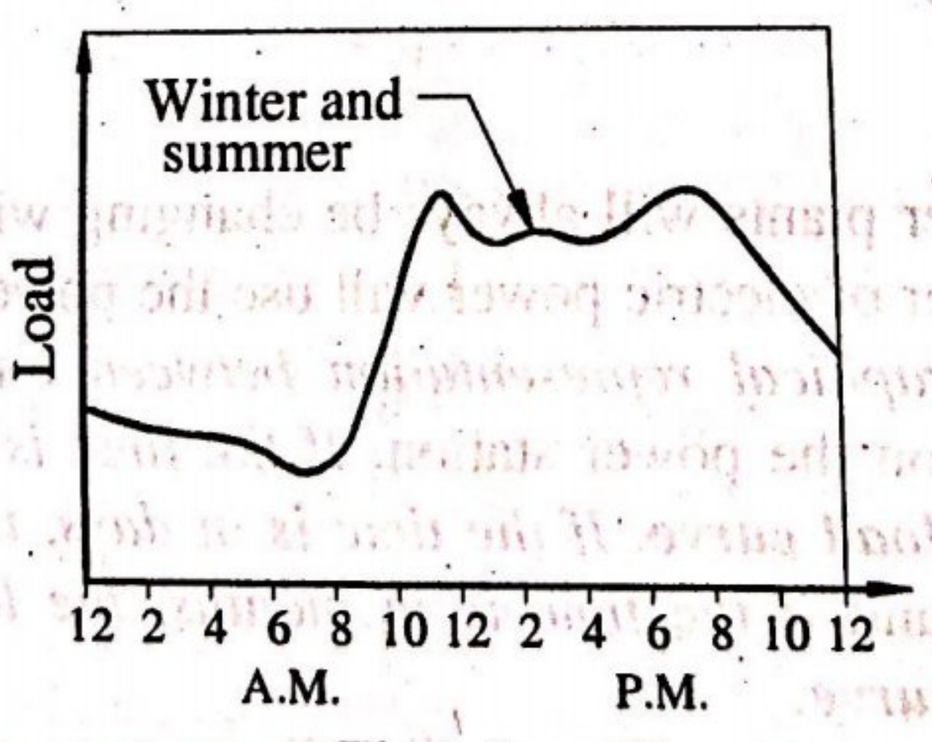
(b) Street lighting load curve



(c) Industrial load curve



(d) Urban traction load curve

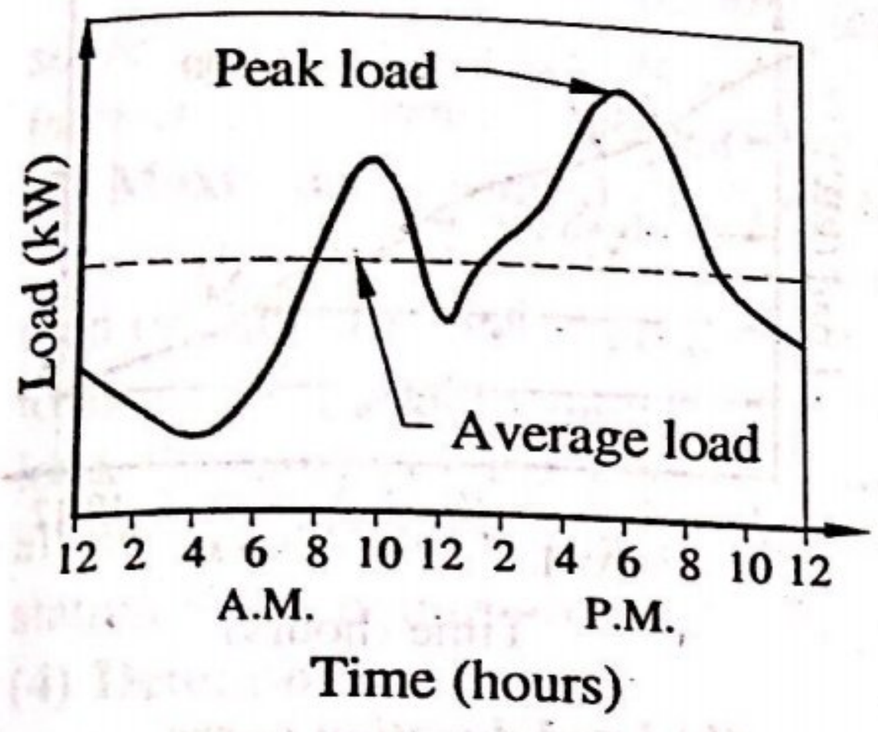


(e) Commercial loads of shops, hospitals, offices etc.

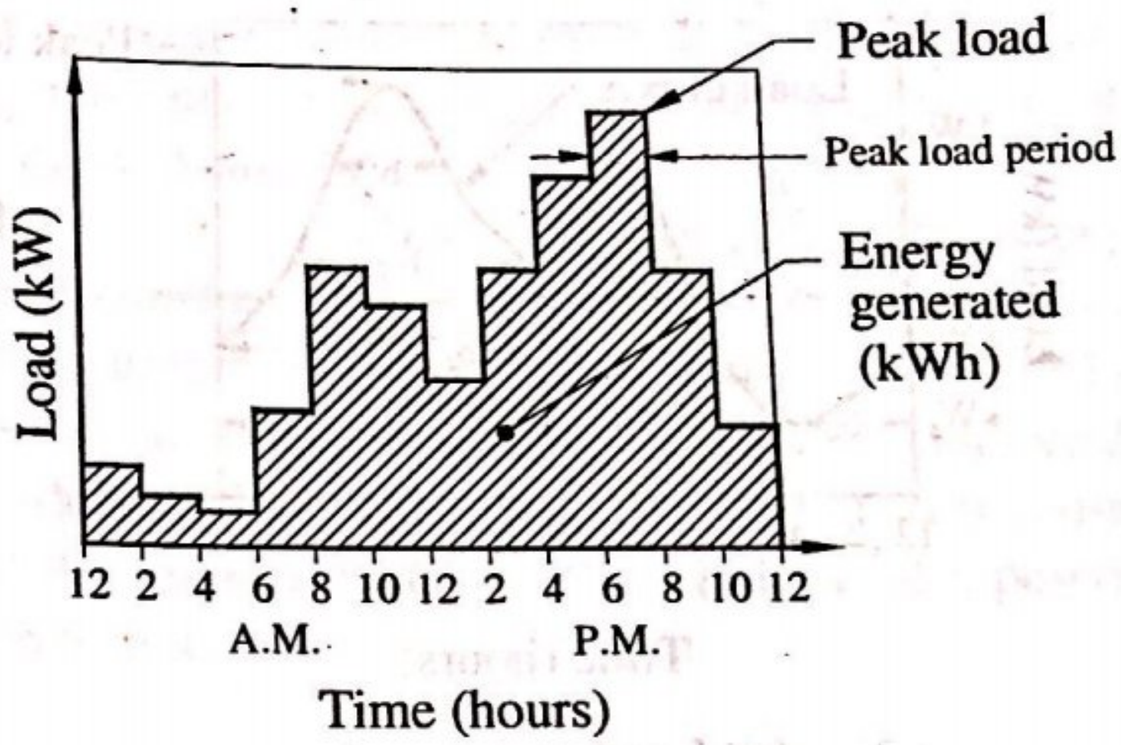
Fig. 12.1 Load curves of different customers

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The combined daily load curve for all types of consumers is shown in Fig. 12.2(a) and the approximated curve for simplicity is shown in Fig. 12.2(b). These curves are called demand curves.



(a) Total load curve (actual)



(b) Theoretical load curve

Fig. 12.2 Daily load curve for all types of consumers.

Following information is available from load curves :

(1) The areas under the load curve represents the energy generated in the period considered.

The energy consumption of the customer is given by an expression,

$$E = \int_0^{24} (kW)\,dt$$

if the load curve is drawn on hourly basis.

where, t is time in hours.

(2) The area under the curve divided by the total number of hours gives the average load on the power station. The peak load indicated by the load curve represents the maximum demand on the power station.

Load curves give full information about the incoming load and help to decide the installed capacity of the power station and to decide the economical sizes of various generating units. They also help to estimate the generating cost and to decide the operating schedule of the power station.

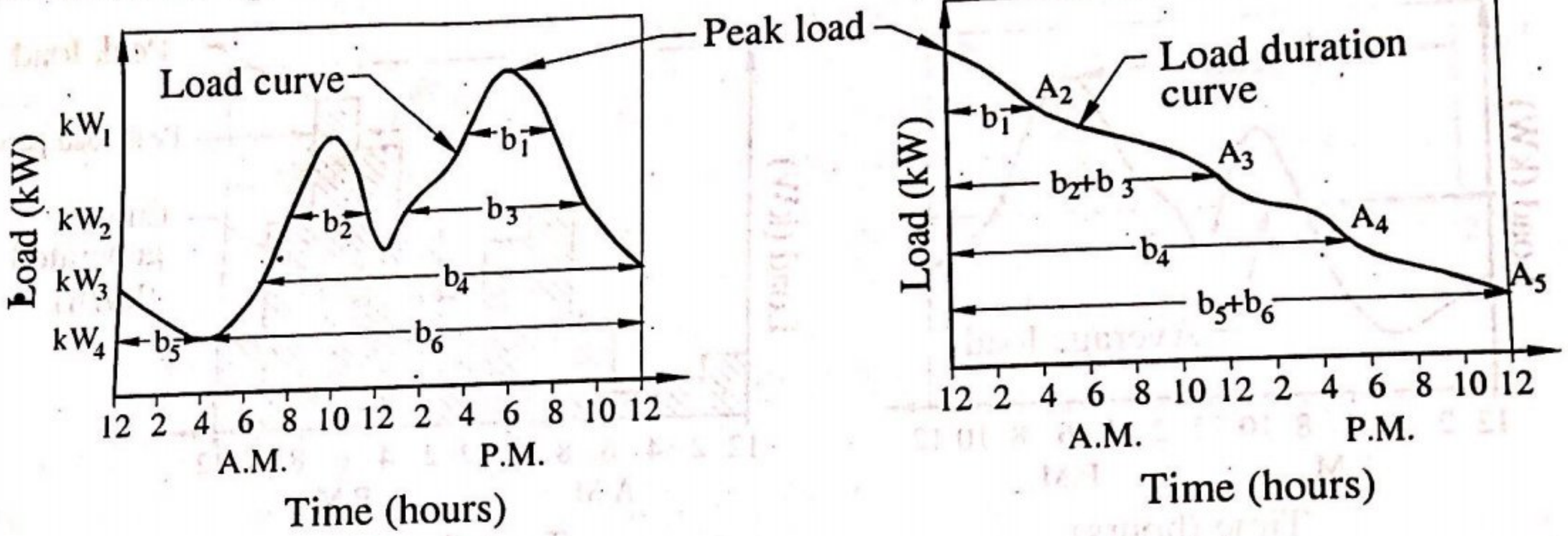
12.3 Load duration curves

Load duration curve is simply a re-arrangement of daily load curve with loads set up in descending order of magnitude. The areas under the load duration curve and corresponding load curve are equal and measure kWh of energy of that period.

The load duration curve indicates for how many hours a certain load is required in a day. To get the load duration curve from chronological load curve, we cut the daily

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load curve into many many vertical strips and then arrange them in descending order. The graphical method for constructing load duration curve from the load curve is described as shown in Fig. 12.3.



(a) Load curve *(b) Load duration curve*

Fig. 12.3 Load curve and load duration curves

The base of the load duration curve represents the total time span *e.g.* 24 hours. The peak load on chronological curve is then plotted as a point at 0 hour on load duration curve. At load ordinate 'kW_1' the intercept 'b_1' of load curve is plotted as time ordinate 'b_1' on the load duration curve as shown. At load ordinate 'kW_2' on load curve the intercepts b_2 and b_3 are plotted on load duration curve as $b_2 + b_3$ and so on. At minimum load A_5 the intercept covers the entire period of 24 hours. The locus of all the points A_1, A_2 etc. represents the daily load duration curve.

12.4 Terms and definitions

(1) Connected load [Dec. '13, June '11, Dec. '11]

Connected load is the sum of ratings in kilowatts (kW) of equipment installed in the consumer's premises. The connected loads in the premises of a consumer are shown in Fig. 12.4.

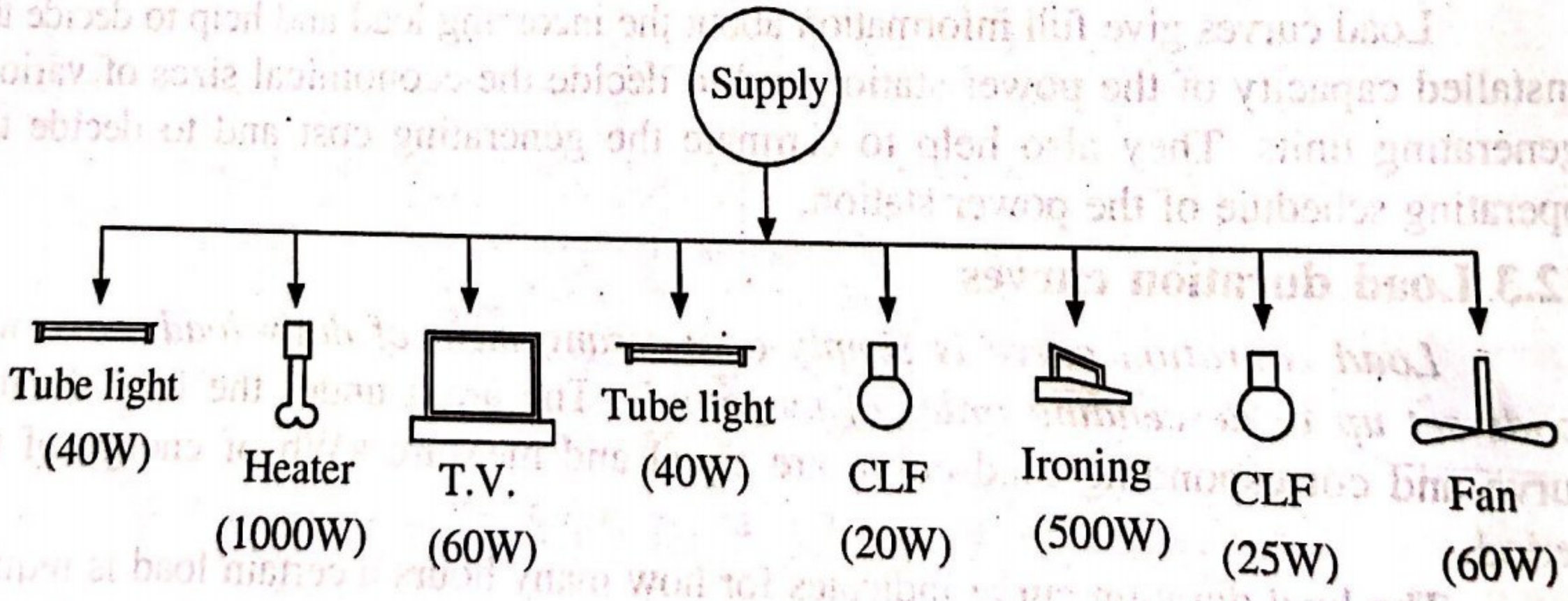


Fig. 12.4 Load in a consumer's premises

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Total load connected in the consumer's premises

$= 40 + 1000 + 60 + 40 + 20 + 500 + 25 + 60 = 1745$ W

(2) Demand :

The demand of an installation or system is the load that drawn from the source of supply at the receiving terminals averaged over a suitable and specified interval of time. Demand is expressed in kilowatts (kW) or other suitable units.

(3) Maximum demand or Peak load **[Dec. '13, June '11, Dec. '11]**

It is the maximum load which a consumer uses at any time. It can be less than or equal to connected load. If all the equipment fitted in consumer's premises run to their fullest extent simultaneously then the maximum demand will be equal to connected load. But generally the actual maximum demand is less than the connected load because all the devices never run at full load at the same time. Maximum demand of a power station is the maximum load on the power station in a given period.

(4) Demand factor :

It is defined as the ratio of maximum demand to connected load.

(5) Average load **[Dec. '13, June '11, Dec. '11]**

Consider load curve shown in Fig. 12.5. *The average load is calculated dividing the area under the load curve (energy in kWh) by the time period (24 hours) considered to draw the load curve.*

$$\therefore \text{Average load} = \frac{\text{Area under load curve}}{24} = \frac{\text{Energy consumed in 24 hours}}{24}$$

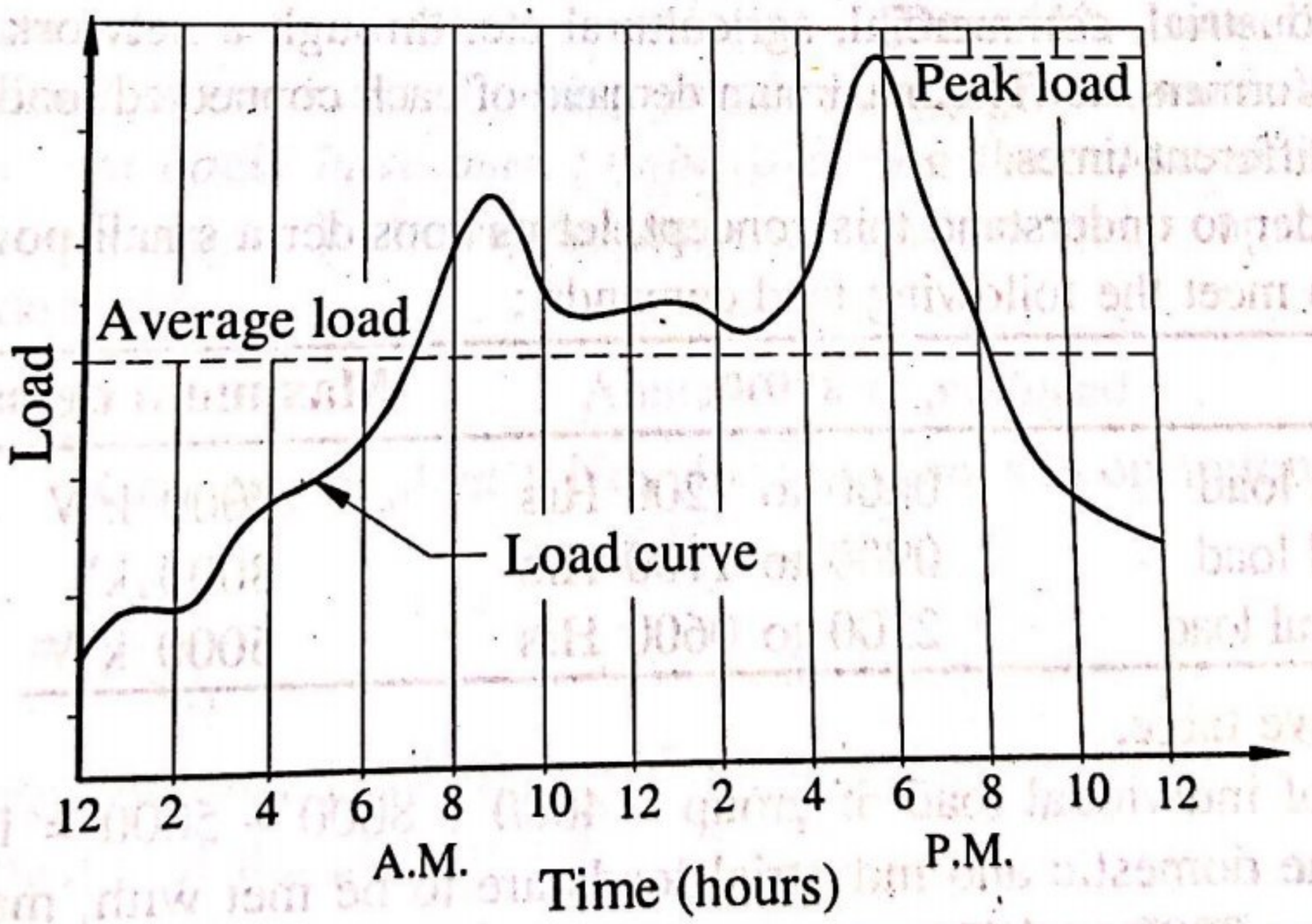


Fig. 12.5 Load curve

Number of units consumed per day are equal to the area under the daily load curve.

Similarly, the monthly average load and yearly average load can be calculated

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with the help of monthly and yearly load curves respectively.

(6) Load factor : **[Dec. '13, June '11, Dec. '11]**

It is defined as the ratio of average load to maximum or peak load. Load factors and demand factors are always less than unity. Load factors play an important part on the cost of generation per unit. The higher the load factor the lesser will be the cost of generation per unit for the same maximum demand.

$$\therefore \text{Load factor} = \frac{\text{Average load}}{\text{Maximum load}}$$

The load factor can also be given by

$$\text{Load factor} = \frac{\text{Total energy consumption in 24 hours}}{\text{Peak load} \times 24}$$

Load factors for different types of consumers are as follows :

(a) Residential load	10-15%
(b) Commercial load	25-30%
(c) Municipal load	25%
(d) Industrial load	
(i) Small scale industries	30-50%
(ii) Medium size industries	55-60%
(iii) Heavy industries	70-80%

(7) Diversity factor : **[Dec. '13, June '11, Dec. '11]**

The power plant supplies generated power to different types of consumers like domestic, industrial, commercial, agricultural etc. through a network of substations, feeders, transformers etc. The maximum demand of each connected load is different and it occurs at different times.

In order to understand this concept, let us consider a small power plant which is required to meet the following load demands :

Load	Time	Maximum demand
(i) Domestic load	0600 to 2200 Hrs	4000 kW
(ii) Industrial load	0900 to 1700 Hrs	8000 kV
(iii) Agricultural load	2300 to 0600 Hrs	5000 kW

From the above table,

Sum of individual load of group = 4000 + 8000 + 5000 = 17000 kW

When both the domestic and industrial loads are to be met with, maximum demand occurs between 0900 to 1700 hrs.

∴ The maximum demand at any point during the day

= 4000 + 8000 = 12000 kW

The ***diversity factor*** *is the ratio of the sum of the maximum demands of the individual consumers and the simultaneous maximum demand of the whole group*

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during a particular time.

$$\therefore \text{Diversity factor} = \frac{\text{Sum of individual maximum demands}}{\text{Simultaneous maximum demand at a given time}}$$

In the above example, the diversity factor is (17000/12000) i.e. 1.4167.

Diversity factor is always greater than unity.

(8) Plant capacity factor : **[Dec. '13, June '11, Dec. '11]**

It is defined as the ratio of actual energy produced in kilowatt hours (kWh) to the maximum possible energy that could have been produced during the same period.

$$\therefore \text{Plant capacity factor} = \frac{\text{Average load} \times 24}{\text{Plant capacity} \times 24}$$

$$= \frac{\text{Average load}}{\text{Plant capacity}} = \frac{\text{Peak load} \times \text{Load factor}}{\text{Plant capacity}}$$

The difference between load and plant capacity factors is an indication of reserve capacity. The capacity factor shows how near the plant runs to its full rating.

If the rated capacity of the plant is equal to the peak load, then the load factor and capacity factor will be numerically equal.

The high values of demand factor, load factor, diversity factor and capacity factor are always desirable for economic operation of the plant and to produce the energy at a cheaper rate.

(9) Plant use factor : **[Dec. '13, June '11, Dec. '11]**

It is defined as the ratio of energy produced in a given time to the maximum possible energy that could have been produced during the actual number of hours the plant was in operation. It shows the extent to which the plant capacity is used to meet the peak demand.

$$\text{Plant use factor} = \frac{\text{Annual energy produced}}{\text{Capacity of plant} \times \text{No. of hours plant is in operation during year}}$$

$$= \frac{E}{C \times t_1}$$

where, E = Energy produced (kWh) in a given period.

C = Capacity of the plant in kW

t_1 = actual number of hours the plant has been in operation.

As the use factor approaches 1, it indicates the need for additional capacity of the plant.

As shown in Fig. 12.6 the different factors are indicated on the diagram.

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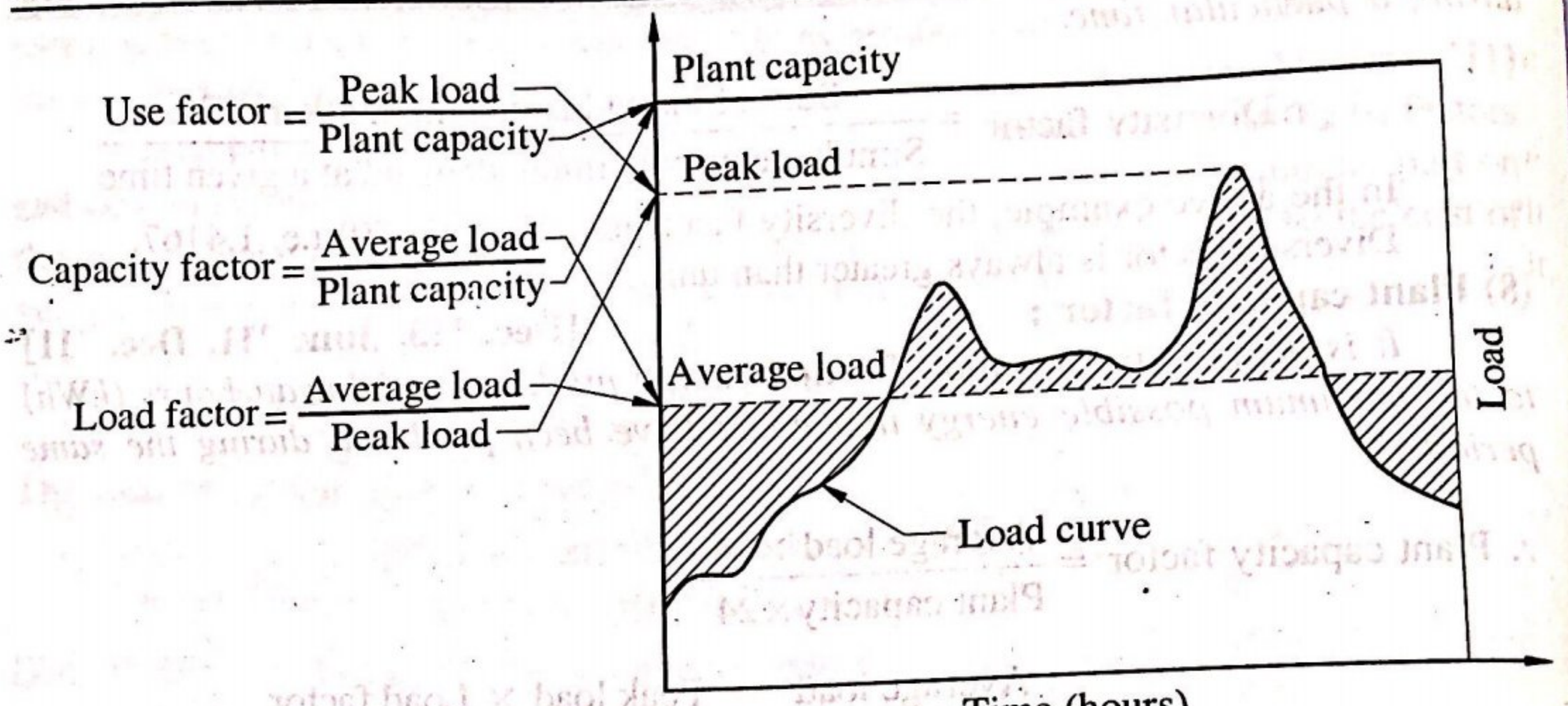


Fig. 12.6 Representation of different factors

12.5 Importance of Load factor and diversity factor

Importance of load factor and diversity factor in cost of power generation is described below.

(1) Load factor

Load factor is the ratio of average load to maximum load on the power plant.

The load factor will increase if the average load increases without the increase in maximum load. Thus, the total number of units of energy generated (kWh) at higher load factor would increase. But the annual fixed charges of power plant remains constant irrespective of total number of units of energy generated. Hence, the annual fixed charges per unit of energy generated would reduce with the increase in load factor. As a result the overall cost per unit of energy generated reduces.

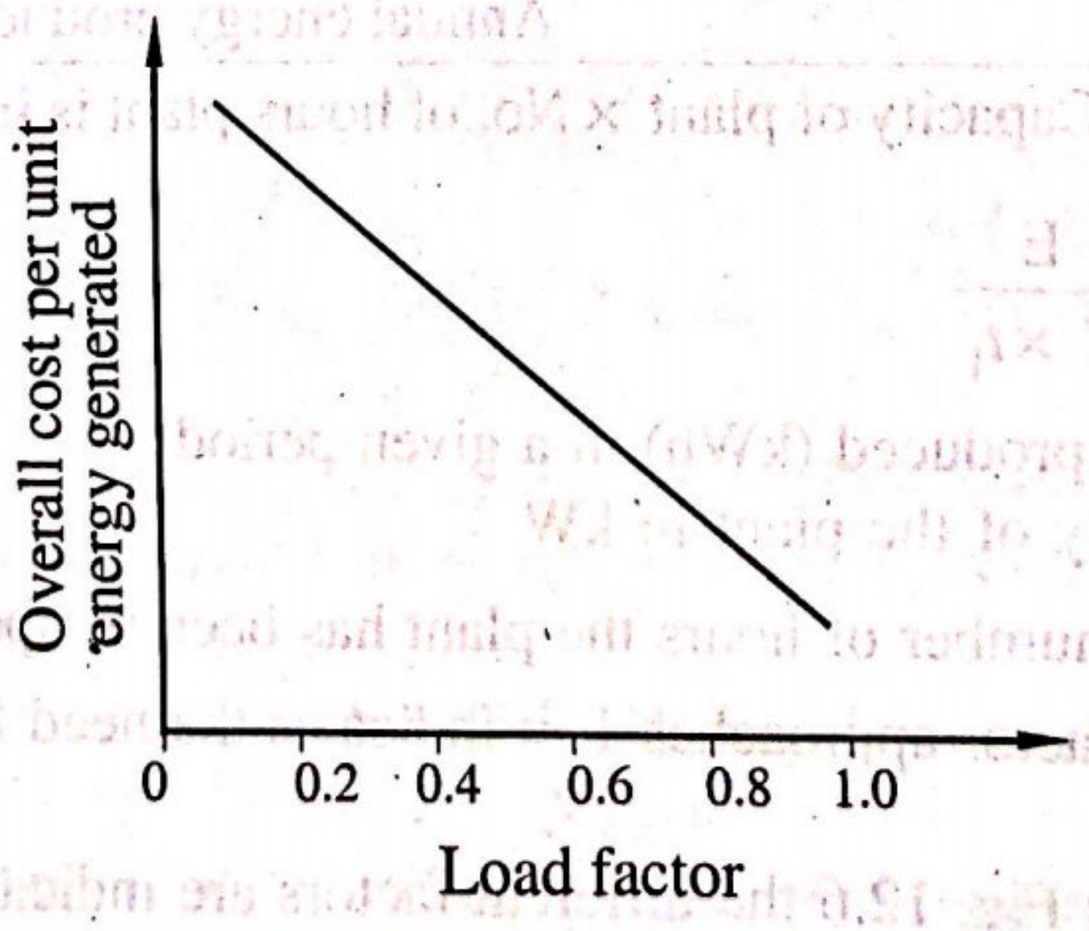


Fig. 12.7 Effect of load factor on cost per unit

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The effect of load factor on overall cost per unit of energy generated is shown in Fig. 12.7.

If the maximum load on plant reduces for the same average load, the load factor will increase. Thus, the kW rating of plant would reduce to meet the demand. Therefore, the improved load factor results into decreased installed capacity of power plant. Hence, it reduces the capital investment on the plant installed.

Low load factor is an indication of high cost of electric production as the power plant is not used to its full capacity for the whole period considered. For the low production cost of the electric energy, it is always desirable to run the plant to its full capacity for the maximum period of operation to give high load factor. Therefore the base load plants (very high capacity) run at high load factor (nearly unity) and peak load plants are allowed to run at low load factors to reduce the overall production cost.

(2) Diversity factor **[May '13]**

The output power of a single power plant is subdivided and supplied to a group of consumers through many feeders from the sub-stations in order to meet their diverse demands

In order to understand the effect of diversity factor, let us consider Fig. 12.8.

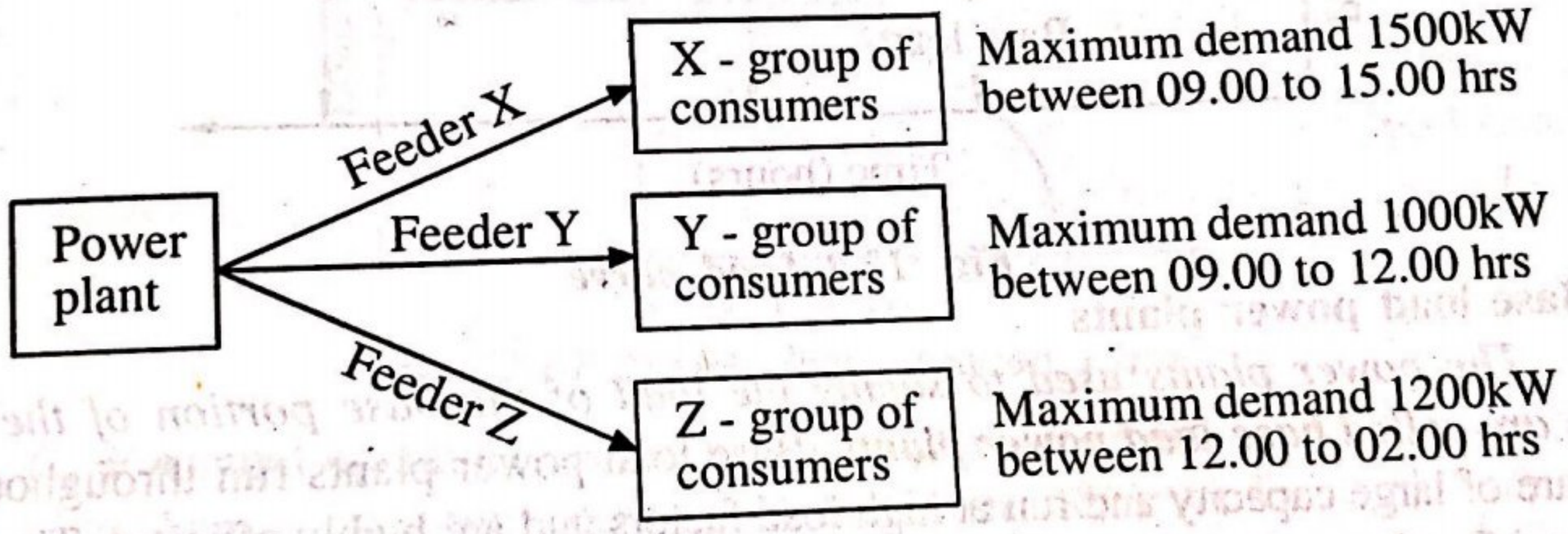


Fig. 12.8 Maximum load demand by various group of consumers

$$\text{Diversity factor} = \frac{\text{Sum of individual maximum demands}}{\text{Simultaneous maximum demands at given time}}$$

$$= \frac{1500+1000+1200}{1500+1000} = 1.48$$

In second case when the maximum demand of Y group of consumers shifts from the period between 0900 to 1200 hrs. to 0300 to 0900 hrs with the same maximum demand of 1000 kW. In this case, the simultaneous maximum demand of group of consumers will come down to 1500 kW. Then,

$$\text{New diversity factor} = \frac{1500+1000+1200}{1500} = 2.4667$$

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From new diversity factor, one can say that the simultaneous maximum demand on the power station reduces with the increase in diversity factor, as a result, the capacity of plant in kW to meet the load demand of the entire system reduces. Therefore, it reduces the capital investment on the plant installed.

From the above, high diversity factor is always desirable for the economic operation of the plant.

12.6 Base load and peak load power plants

The power plants work at different load factors. A typical load curve is shown in Fig. 12.9.

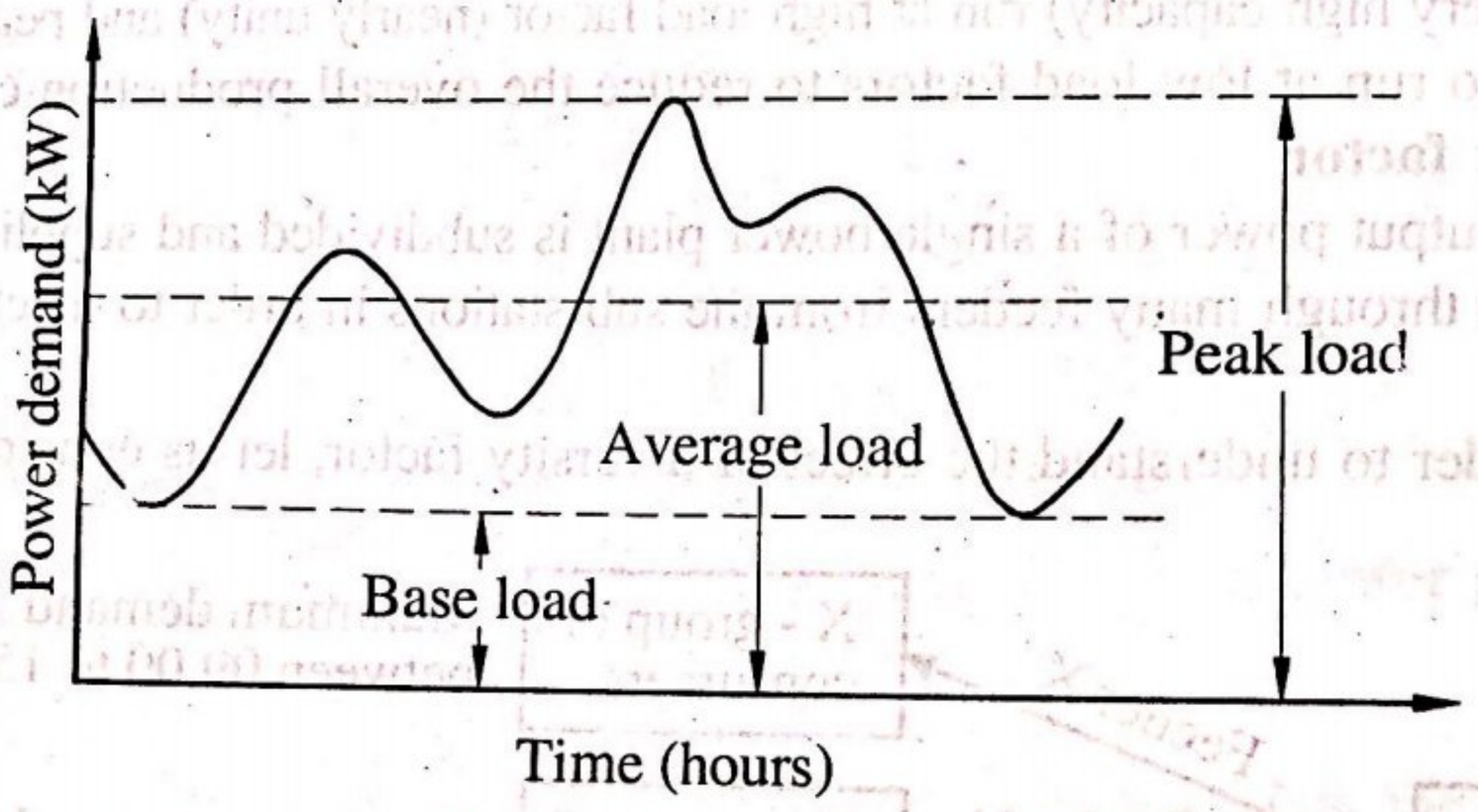


Fig. 12.9 Load curve

(1) Base load power plants

The power plants used to supply the load of the base portion of the load curve are called ***base load power plants***. Base load power plants run throughout the year, are of large capacity and run at high load factors and are highly efficient. The fixed and semi-fixed cost of these plants is usually high. Thermal, nuclear and hydro plants are generally considered as base load plants. The requirements of a base load power plant are as follows :

(1) Its operation cost should be low as it has to operate most of the time.
(2) It should be able to supply the load continuously.
(3) Its maintenance cost should be low.

(2) Peak load power plants

The power plants which supply the load on the top portion of load curve are called ***peak load plants***. They are for use during peak loads. Accordingly, they have to be started up and shut down several times a day. They are of smaller capacity, run for a short period and work at low load factors. Peak load plants should be capable of quick starting. The peak load plants adopted in practice are gas turbine, diesel engine and simple steam turbine plants. The requirement of a peak load plant are as follows :

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(1) It should be capable of being started from cold conditions within minimum time.
(2) Its capital cost should be minimum.
(3) Its operating cost should be low as possible.

Parallel operation of different power stations and the co-ordination of electricity generation leads to considerable saving in comparison with the same load fed by independent power stations.

12.7 Economic load sharing between base load and peak load power stations

Consider a system having load duration curve shown in Fig. 12.10. The load to be supplied by a base load power plant and peak load power plant. It is possible to find out the most economic load sharing between them if the operating costs of base load and peak load plants are known.

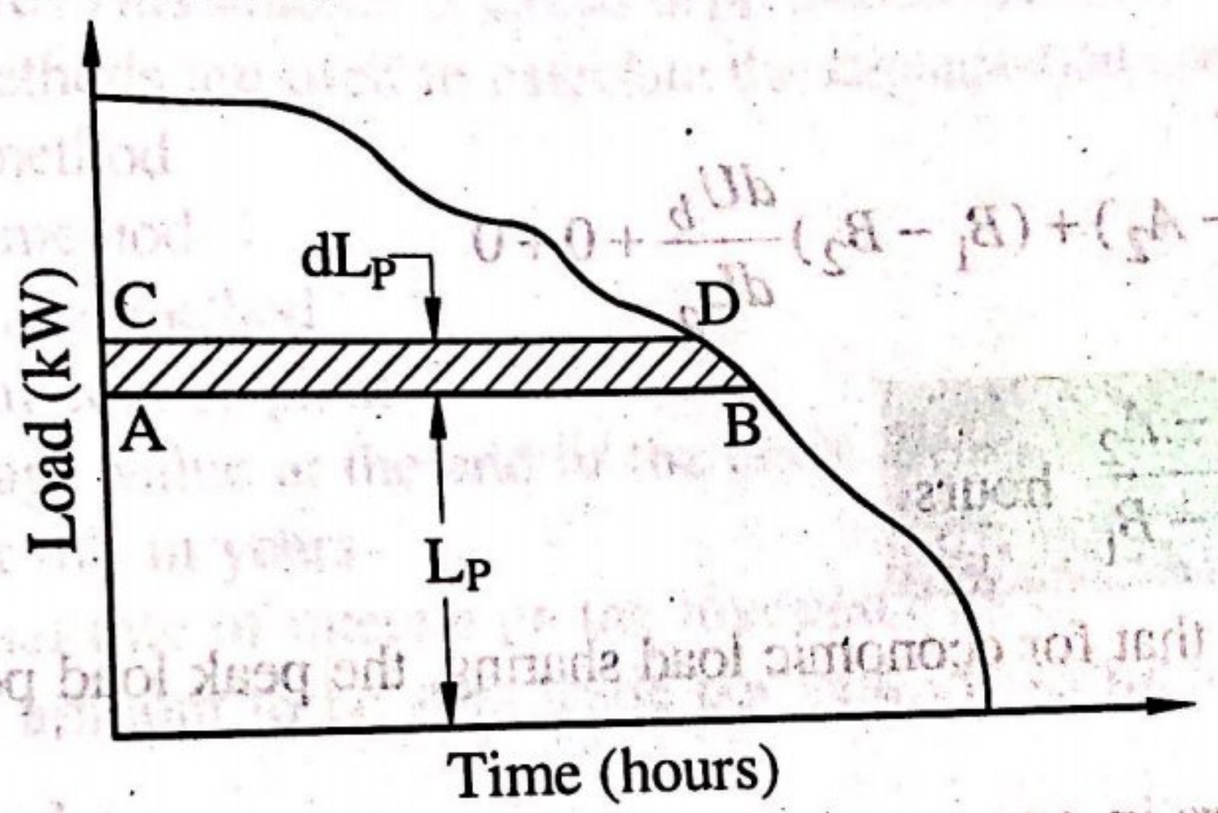


Fig. 12.10 Annual load duration curve

Let C_1 = Annual operating cost of base load plant (Rs./kW)
C_2 = Annual operating cost of peak load plant (Rs./kW)
A_1 = Fixed cost of base load plant (Rs./kW)
B_1 = Variable cost of base load plant (Rs/kWh)
A_2 and B_2 are the corresponding values of peak load plant.
L_p = Peak load on base load plant
L = Peak load on the system
B = Base load on the system
U = Total number of unit (kWh) generated.
U_b = Total number of unit generated by base load plant (kWh)

Assuming $A_1 > A_2$ and $B_1 < B_2$.

Annual operating cost of base load plant

$$C_1 = A_1\,kW + B_1\,kWh$$

$$\therefore C_1 = A_1 L_p + B_1 U_b$$

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Annual operating cost of peak load plant,

$$C_2 = A_2\,kW + B_2\,kWh$$

$$= A_2(L - L_P) + B_2(U - U_b)$$

Total annual cost of power generation; C,

$$C = C_1 + C_2$$

$$= [A_1 L_P + B_1 U_b] + [A_2(L - L_P) + B_2(U - U_b)]$$

$$= (A_1 - A_2)L_p + (B_1 - B_2)U_b + A_2 L + B_2 U$$

Now for minimum cost

$$\frac{dC}{dL_p} = 0$$

This gives, $0 = (A_1 - A_2) + (B_1 - B_2)\frac{dU_b}{dL_p} + 0 + 0$

$$\therefore \boxed{\frac{dU_b}{dL_p} = \frac{A_1 - A_2}{B_2 - B_1}\ \text{hours}} \qquad ...(12.1)$$

This indicates that for economic load sharing, the peak load power station should run for $\frac{A_1 - A_2}{B_2 - B_1}$ hours in one year.

12.8 Cost of power plant [Jan. '13]

The cost analysis of power plant includes fixed cost and running cost.

(1) Fixed cost

The fixed cost is the capital invested in the installation of complete plant. This cost includes the following :

(i) Land cost
(ii) Building cost
(iii) Equipment cost
(iv) Installation cost
(v) cost of transmission and distribution lines
(vi) cost of planning and designing the plant
(vii) depreciation cost, interest on the capital invested insurance, etc.

(i) Land building and equipment cost :

Cost of the land and building will depend upon the location of the plant. If the plant is situated near the cities, the land will be costlier than the case if it is located away from the cities. The cost of the equipment or the plant investment cost is usually expressed on the basis of kW capacity installed.

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(ii) Interest :

All enterprises need investment of money and this money may be obtained as loan, through bonds and shares, or from owners of personal funds. The interest on the capital investment must be considered because otherwise if the same amount was not invested in power plant, it would have earned an annual interest. A suitable rate of interest must be considered on the capital invested.

(iii) Depreciation cost :

Depreciation accounts for the deterioration of the equipment and decrease in its value due to corrosion, weathering, and wear and tear with use. It also covers the decrease in value of equipment due to obsolescence. It is required to replace the generating plant machinery after its expiry of useful life. Therefore, a certain amount is kept aside every year from the income of the plant to enable the replacement of plant at the end of its useful life. This amount is called depreciation amount.

The following methods are used to calculate the depreciation cost :

(a) Straight line method

(b) Sinking fund method

(c) Diminishing value method

Let P = Initial cost of plant

S = Salvage value at the end of the plant life.

n = Plant life in years

r = Annual rate of interest on the invested capital

A = The amount to be kept aside per year as depreciation amount

(a) Straight line method :

It is the simplest and commonly used method. In this method, annual amount to be set aside is calculated by using straight line law.

$$\therefore A = \frac{P - S}{n} \quad \ldots(12.2)$$

In this method the rate of depreciation is uniform throughout the life of the equipment. The depreciation amount is independent on the rate of interest it may draw.

(b) Sinking fund method :

In this method, the amount set aside per year consists of annual installments and the interest earned on all the installments.

Depreciation amount set aside at the end of first year = A

Depreciation amount at the end of second year = A + interest on A

$= A + Ar = A(1 + r)$

Depreciation amount at the end of third year

= $A(1 + r)$ + interest on $A(1 + r)$

= $A(1 + r) + A(1 + r)r = A(1 + r)^2$

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$\therefore$ Amount at the end of nth year $= A(1+r)^{n-1}$

Total amount accumulated in n years

= Sum of the amounts accumulated in n years

$$P - S = A + A(1+r) + A(1+r)^2 + \ldots + A(1+r)^{n-1}$$

$$y = A[1 + (1+r) + (1+r)^2 + \ldots + (1+r)^{n-1}] \quad \ldots(12.3)$$

($\because$ Taking $P - S = y$)

Multiplying the above equation by $(1 + r)$, we get

$$y(1+r) = A[(1+r) + (1+r)^2 + (1+r)^3 + \ldots + (1+r)^n] \quad \ldots(12.4)$$

Subtracting equation (12.3) form equation (12.4), we get

$$y \cdot r = [[(1+r)^n - 1]A$$

$$\therefore y = \left[\frac{(1+r)^n - 1}{r}\right]A$$

$$\therefore (P - S) = \left[\frac{(r+1)^n - 1}{r}\right]A$$

$$\therefore A = \left[\frac{r}{(1+r)^n - 1}\right](P - S) \quad \ldots(12.5)$$

(c) Diminishing value method :

In this method the deterioration in value of equipment from year to year is taken into account and the amount of depreciation calculated upon actual residual value for each year. It thus, reduces for successive years.

Let x % of amount is set aside per year on the initial cost of plant at the end of each successive years.

Initial cost of the plant $= P$

Depreciation amount at the end of first year $= P \times \frac{x}{100}$

Balance plant cost $= P - \frac{P \cdot x}{100} = P\left(1 - \frac{x}{100}\right)$

Depreciation amount at the end of second year,

$$= P\left(1 - \frac{x}{100}\right) \cdot \frac{x}{100}$$

Balance plant cost at the end of second year

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$$= P\left(1-\frac{x}{100}\right) - P\left(1-\frac{x}{100}\right)\left(\frac{x}{100}\right)$$

$$= P\left(1-\frac{x}{100}\right)^2$$

Depreciation amount at the end of third year,

$$= P\left(1-\frac{x}{100}\right)^2 \cdot \left(\frac{x}{100}\right)$$

∴ Depreciation amount at the end of nth year

$$= P\left(1-\frac{x}{100}\right)^{n-1} \cdot \frac{x}{100} \quad \ldots(12.6)$$

This method requires heavy installments in the early years when the maintenance charges are minimum and it goes on decreasing as the time passes but the maintenance charges increase. This is the disadvantages of this method.

(iv) Insurance :

The costly equipment and the buildings must be insured for the fire risks, riots etc. A fixed sum is set aside per year as insurance charges. The insurance charge depends upon the initial cost of the plant and the insurance coverage.

(v) Management cost :

This includes the salaries of management, security and administrative staff, etc. working in the plant. This must be paid whether the plant is working or not. Therefore, this is included in fixed charges of the plant.

(2) Running cost :

The running cost or operating cost of the power plant includes the cost of fuel, cost of lubricating oil, direct labour cost, cooling water and number of consumable articles required. The wages required for supplying the above material are also included in the operating cost of the power plant.

(i) Fuel cost :

In a thermal power plant, fuel is the heaviest item of operating cost. The selection of the fuel and the maximum economy in its use are, therefore, very important consideration in thermal power plant design. The cost of fuel includes not only its price at the site of purchase but its transportation and handling cost also.

(ii) Oil, Grease and Water cost :

The cost of various consumables like oil, grease, etc. and water cost are also proportional to the amount of power generated. These costs increase with an increase in life of the plant as the efficiency of the power plant decreases with the age.

The total cost of power generated is the sum of fixed charges and operating charges.

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12.9 Performance and operating characteristics of power plant

The performance of generating power plants is compared by their average efficiency over a period of time. *The average efficiency of a power plant is the ratio of useful energy output to the total energy input during the period considered.*

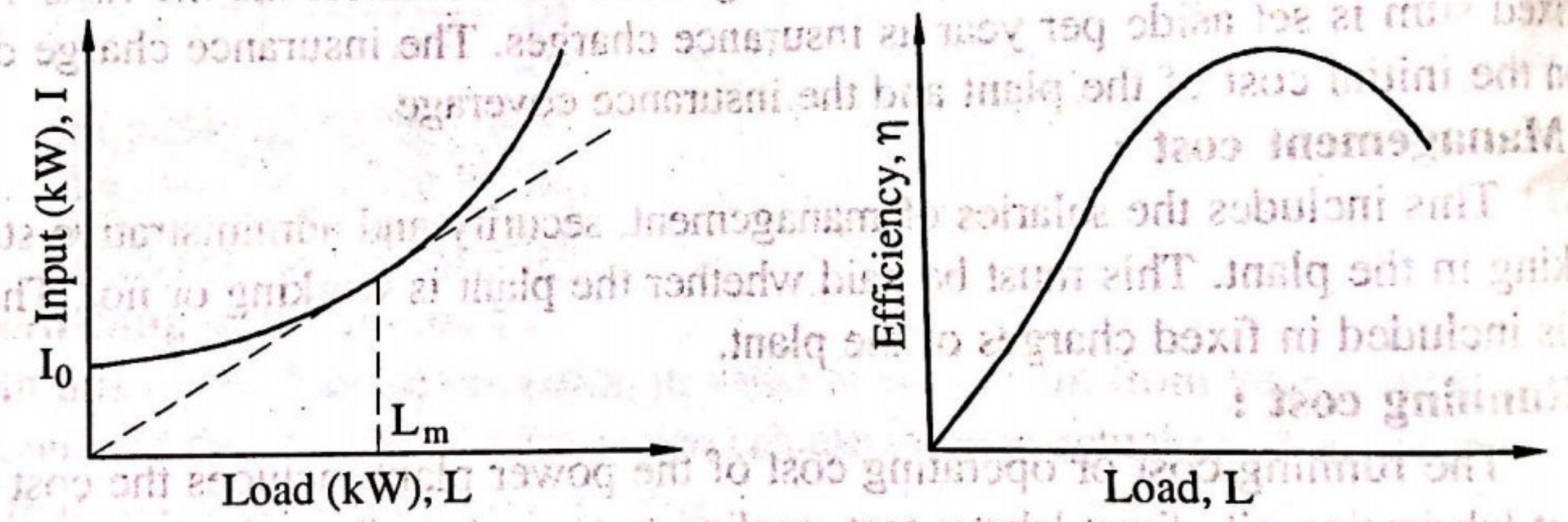
The performance of a plant can be precisely represented by the input-output curve from the tests conducted on individual power plant. The input-output curve is graphical representation between the net energy output (L) and input (I). The input is generally expressed in MJ/h and load output is expressed as megawatts (MW). The input to hydro-plant is measured is m^3/s.of water.

In general input-output may be represented as follows :

$$I = a + bL + cL^2 + dL^3 \quad \ldots (12.9)$$

where a, b, c, and d are constants.

• **Input-output curve :** Fig. 12.11(a) shows an input-output curve. In order to keep the plant in running condition at zero load, a certain input (I_0) is required to meet frictional and heat losses.



(a) Input-output curve *(b) Efficiency curve*

Fig. 12.11

• **Efficiency curve :** *The efficiency of the power plant is defined as a ratio of output to input.*

$$\therefore \eta = \frac{L}{I} = \frac{L}{a + bL + cL^2 + dL^3}$$

The efficiency curve is shown in Fig. 12.11(b).

• **Heat rate and incremental rate curves**

The heat rate is defined as the input per unit output.

$$\therefore \text{Heat Rate (HR)} = \frac{I}{L} \frac{kJ}{kWh}$$

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$$= \frac{a + bL + cL^2 + dL^3}{L} = \frac{a}{L} + b + cL + dL^2$$

The heat rate curve is derived by taking at each load the corresponding input. This curve is drawn as shown in Fig. 12.12.

The relation between efficiency and heat rate is given by

$$\eta = \frac{1}{HR}$$

- **Incremental Rate (I.R.)** *is defined as the ratio of additional input (dI) required to increase the additional output (dL).* Therefore

$$IR = \frac{dI}{dL} \frac{kJ}{kWh}$$

$$\therefore dI = IR \times dL$$

$$\therefore \int_{I_1}^{I_2} dI = \int_{L_1}^{L_2} IR\, dL$$

$$\therefore I_2 - I_1 = \int_{L_1}^{L_2} IR\, dL$$

This indicates that the area under the incremental rate curve gives the total input to the plant to increase the output from L_1 to L_2 as shown in Fig. 12.12.

The total input from no-load to full load is given by

$$I_f - I_o = \int_{L_o}^{L_f} IR\, dL$$

Substituting the value of I in the equation of IR

$$IR = \frac{dI}{dL} = \frac{d}{dL}(a + bL + cL^2 + dL^3) = b + 2cL + 3dL^2$$

This is incremental rate curve equation.

The incremental load curve can be considered a straight line for relatively small increase in output. This is not true for large increment of load because of the marked curvature of the incremental rate curve.

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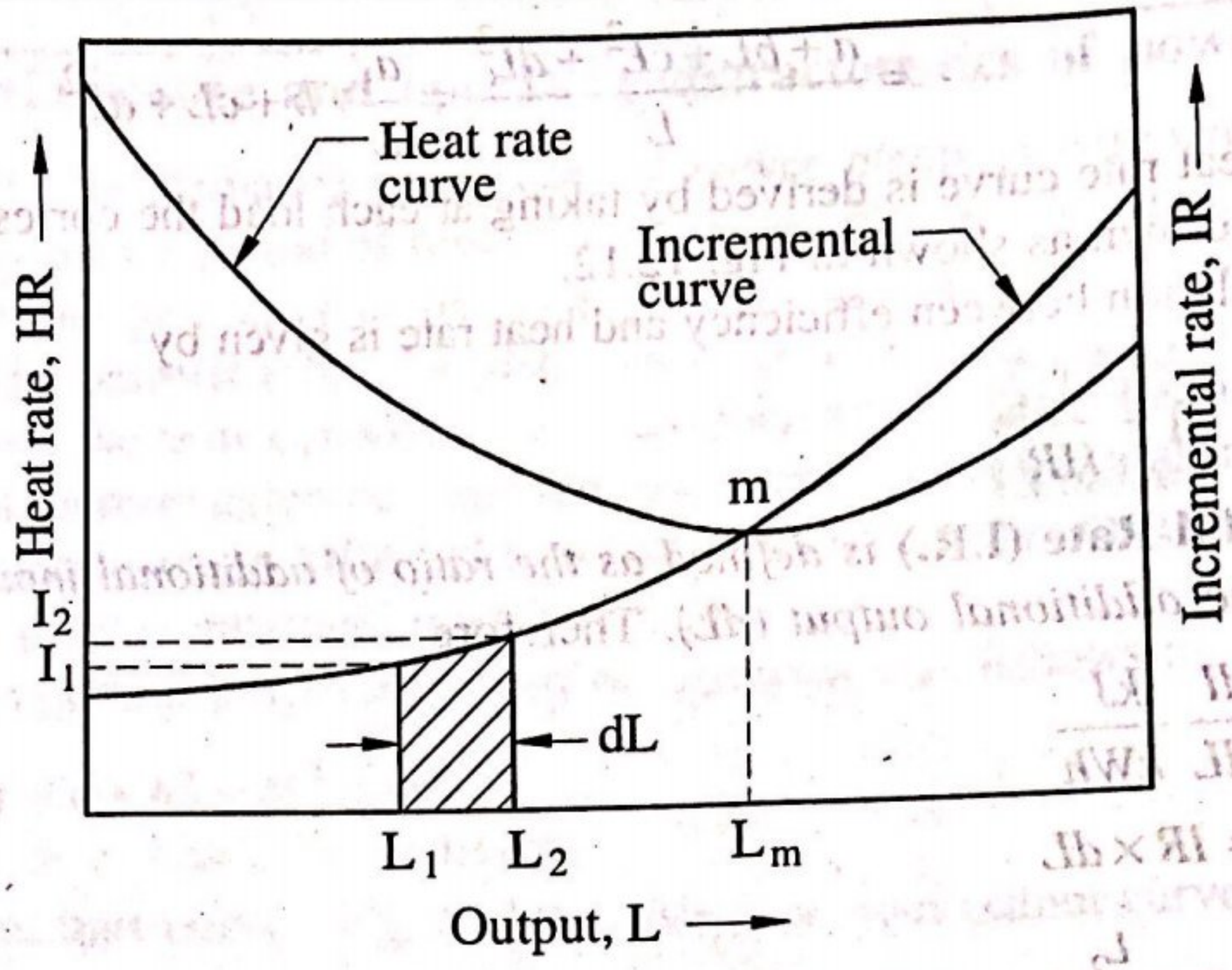


Fig. 12.12 HR and IR curves

From Fig. 12.12, it is evident that the IR curve crosses the HR curve at the lowest value of heat rate. The reason for this can be explained mathematically as given below.

At minimum heat rate, the slope of the heat rate curve must be zero.

$$\therefore \frac{d(HR)}{dL} = \frac{d(I/L)}{dL} = 0$$

$$\therefore \frac{L\,dI - I\,dL}{L^2} = 0$$

$$\therefore L\,dI - I\,dL = 0$$

$$\therefore \frac{dI}{dL} = \frac{I}{L}$$

$\therefore IR = HR$ When HR is minimum

This equation indicates that the heat rate of continuous input-output curve is minimum when it equals the incremental rate. The incremental rate curve has a continuous increasing characteristic, therefore two curves must cross at load L_m when $IR_m = HR_m$.

12.10 Tariff for electric energy

The electricity generated is to be supplied to the consumers, the total cost of generation and profit as to be recovered from the consumers. The rates of energy sold to the consumers depend on the type of consumers as domestic, commercial and

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industrial. The rates depend upon the total energy consumed and the load factor of the consumer. The tariff (energy rates) chosen should recover the fixed cost, operating cost and profit etc. incurred in generating the electrical energy.

The various types of tariffs can be derived from the following general equation :

$$z = ax + by + c$$

where z = Total amount of bill for the period considered.

x = Maximum demand in kW

y = Energy consumed in kW.h during the period considered.

a = Rate per kW of maximum demand.

b = Energy rate of kWh.

c = Constant amount charged to the consumer during each billing period. This charge is independent of demand or total energy because a consumer that remains connected to the line incurs expenses even if he/she does not use energy.

Various type of tariffs are as follows :

(1) Flat demand rate :

It is based on the connected load and fixed number of hours of use per month or year. The rate is expressed as a certain price per kW of connected load of the consumer. This energy rate eliminates the use of metering equipment. It is expressed by the expression

$$z = ax$$

The variation in total cost and unit cost are shown in Fig. 12.13(a)

(2) Straight line meter rate :

According to this energy rate, the amount to be charged from the consumer depends upon the energy consumed in kWh which is recorded by a means of a kilowatt hour meter. It is expressed in the form

$$z = by$$

The drawback of this system, the consumer using no energy will no pay any amount although he/she has incurred some expenses to the power station. Secondly since the rate per kWh is fixed, this tariff does not encourage the consumer to use more power. The variation of bill according to the variation of energy consumed is shown in Fig. 12.13(b).

(3) Block-Meter rate :

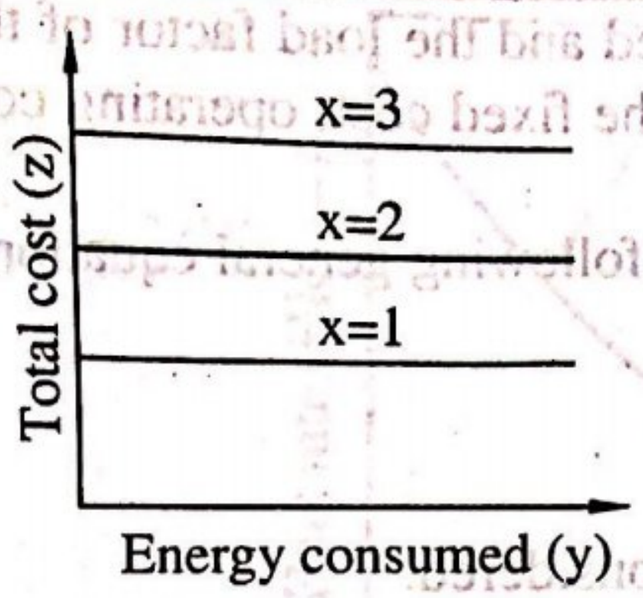
In block-meter rate method, different rates are charged for different blocks of consumption. The size of the block will depend upon the size of the consuming body. With increased consumption of electric energy, the unit rate will go on decreasing. This tariff is expressed by the expression

$$z = b_1 y_1 + b_2 y_2 + b_3 y_3 + \dots$$

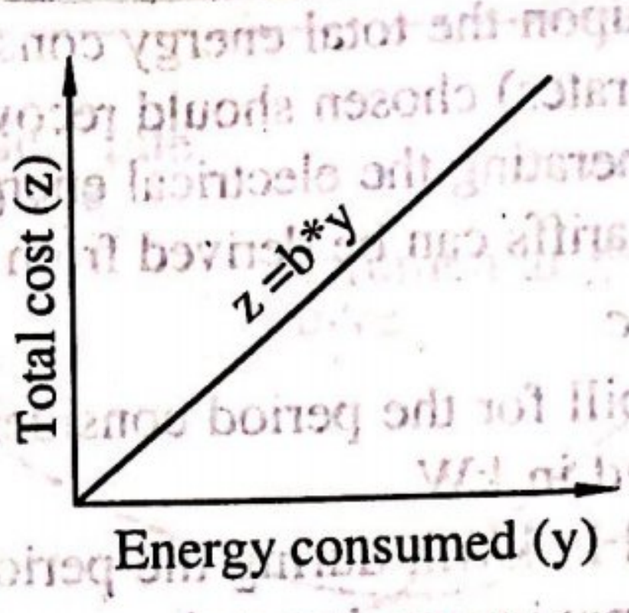
where $b_3 < b_2 < b_1$ and $y_1 + y_2 + y_3 + \dots = y$ (total energy consumption)

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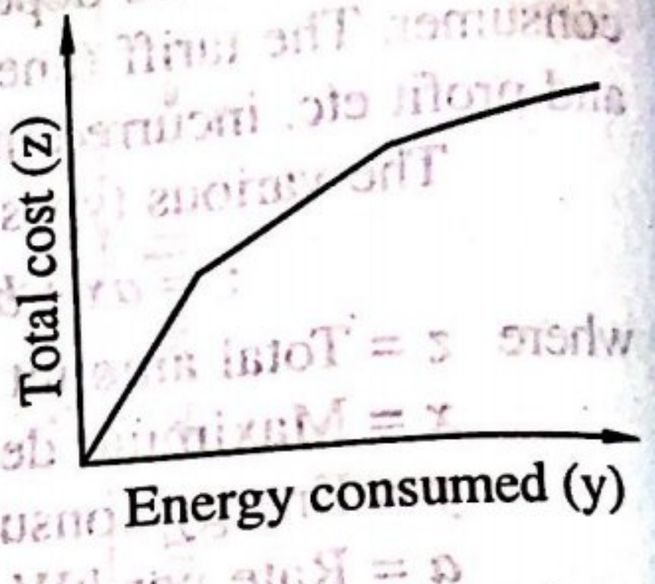
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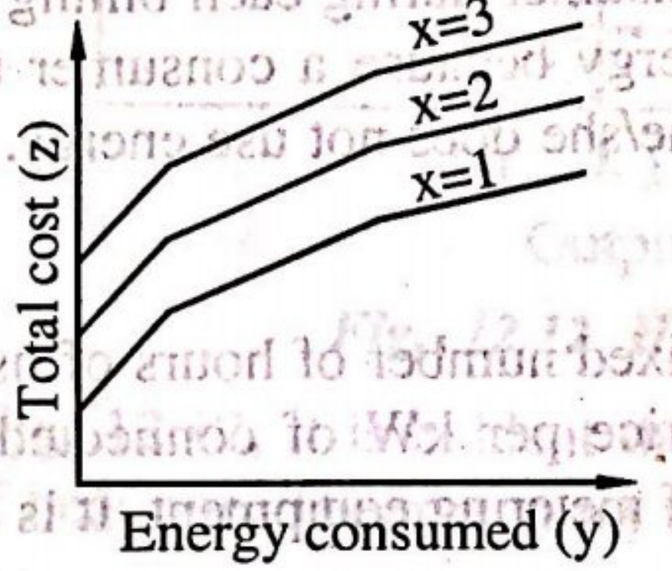
(a) Flat demand rate



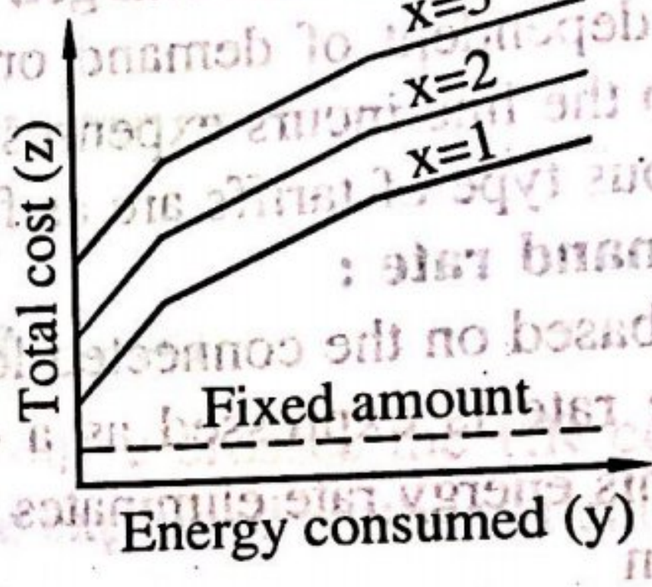
(b) Straight meter rate



(c) Block meter rate



(d) Hopkinson rate



(e) Doherty rate

Fig. 12.13 Basic forms of energy tariff

The level of block y_1, y_2, y_3 and so on is decided by the management to recover the capital cost of the plant. The variation of bill according to this method is shown in Fig. 12.13(c).

(4) Two-part tariff or Hopkinson demand rate :

This method of charging was proposed by Dr. John Hopkinson in 1892. In this tariff the total charges are based on the maximum demand and energy consumed. It is expressed as

$$z = ax + by$$

This method requires two metres to record the maximum demand and the energy consumption of the consumer. The variation of bill according to this method is shown in Fig. 12.13(d).

(5) Three part tariff or Doherty rate :

This method of charging was proposed by Henry L. Doherty. According to this tariff the consumer pays some fixed amount in addition to the charges for maximum demand and energy consumed. The fixed amount to be charged depends upon the occasional increase in fuel price, rise in wages of labour etc. It is expressed by the expression.

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$$z = ax + by + c$$

The variation of bill according to this method is shown in Fig. 12.13(e).

When the generating capacity is less than the actual demand then the customers are discouraged to use more power. Upto the certain power consumption, the charging rate is fixed say 1.5 Rs./kWh upto 50 kWh units and if it exceeds than this, the charge is rapidly increased as 2 Rs./kWh. This is unfortunate but very common in India.

Problem 12.1 *A power station supplies the following loads to the consumers :*

Time : in hours	*0-6*	*6-10*	*10-12*	*12-16*	*16-20*	*20-22*	*22-24*
Load in MW	*20*	*60*	*80*	*50*	*90*	*60*	*50*

(i) Draw the load curve and estimate the load factor of the plant, (ii) What is the load factor of a standby equipment of 20 MW capacity if it takes up all loads above 70 MW ? What is its use factor ?

Solution : Given data

Peak load = 90 MW, capacity of standby unit = 20 MW

Load factor = ?, use factor = ?

(i) The load curve is drawn in Fig. 12.14

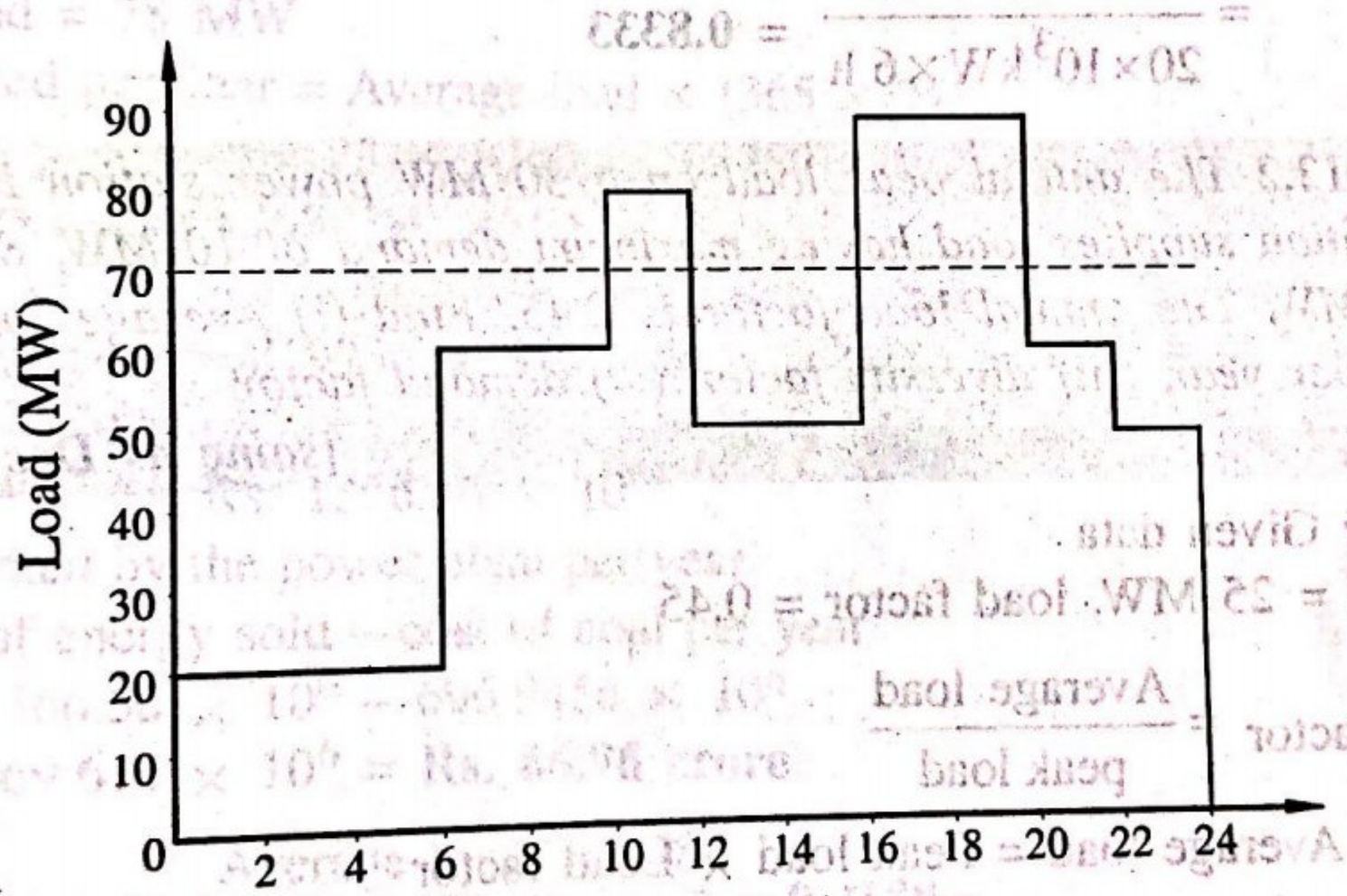


Fig. 12.14.

Energy generated = Area under the load curve

$$= 20\times6 + 60\times4 + 80\times2 + 50\times4 + 90\times4 + 60\times2 + 50\times2$$

$$= 1300 \text{ MWh}$$

$$\text{Average load} = \frac{1300 \text{ MWh}}{24 \text{ h}} = 54.1667 \text{ MW}$$

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$$\text{Load factor} = \frac{\text{Average load}}{\text{peak load}} = \frac{54.1667}{90} = \mathbf{0.6018}$$ **Ans**

(ii) If the load above 70 MW is supplied by a standby unit of 20 MW capacity, the energy generated by it

$$= 10 \times 2 + 20 \times 4 = 100 \text{ MWh}$$

Time during which the standby unit remains in operation

$$= 2 + 4 \text{ (from the load curve)} = 6\ h$$

$$\text{Average load} = \frac{100\,\text{MWh}}{6\,\text{h}} = 16.6667 \text{ MW}$$

$$\text{Load factor} = \frac{\text{Average load}}{\text{peak load}} = \frac{16.6667}{20} = \mathbf{0.8333}$$ **Ans**

$$\text{Use factor} = \frac{\text{Energy generated}}{\text{Plant capacity} \times \text{operating hours}}$$

$$= \frac{100 \times 10^3 \text{ kWh}}{20 \times 10^3 \text{ kW} \times 6 \text{ h}} = \mathbf{0.8333}$$ **Ans**

Problem 12.2 *The annual peak load on a 30 MW power station is 25 MW. The power station supplies load having maximum demand of 10 MW, 8.5 MW, 5 MW and 4.5 MW. The annual load factor is 0.45. Find (i) Average load, (ii) energy supplied per year, (iii) diversity factor, (iv) demand factor.*

[same as Dec '11, 7 marks]

Solution : Given data

Peak load = 25 MW, load factor = 0.45

$$\text{(i) Load factor} = \frac{\text{Average load}}{\text{peak load}}$$

$$\therefore \text{ Average load} = \text{Peak load} \times \text{Load factor}$$

$$= 25 \times 0.45 = 11.25 \text{ MW} = \mathbf{11.25 \times 10^3 \text{ kW}}$$ **Ans**

(ii) Energy supplied per year = Average load $\times$ (365 $\times$ 24)

$$= 11.25 \times 10^3 \times 8760 \text{ kWh}$$

$$= \mathbf{98.55 \times 10^6 \text{ kWh}}$$ **Ans**

$$\text{(iii) Diversity factor} = \frac{\text{Sum of individual maximum demands}}{\text{Simultaneous maximum demand}}$$

$$= \frac{10 + 8.5 + 5 + 4.5}{25} = \frac{28}{25} = \mathbf{1.12}$$ **Ans**

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(iv) Demand factor $= \dfrac{\text{Maximum demand}}{\text{connected load}} = \dfrac{25}{10+8.5+5+4.5} = \dfrac{25}{28}$

$= \mathbf{0.8928}$ ***Ans***

Problem 12.3 *A thermal power plant of 170 MW capacity has the maximum load of 130 MW. Its annual load factor is 0.6. The coal consumption is 1.2 kg per kWh of energy generated and the cost of coal is Rs. 850 per tonne. Calculate (i) the annual revenue earned if energy is sold at Rs. 2 per kWh and (ii) the capacity factor of the plant.*

Solution : Given data

Peak load = 130 MW, load factor = 0.6, coal consumption = 1.2 kg/kWh, coal cost = Rs. 850/tonne, annual revenue = ? Capacity factor = ?

$$\text{Annual load factor} = \frac{\text{Average load}}{\text{peak load}}$$

$$0.6 = \frac{\text{Average load}}{130}$$

∴ Average load = 78 MW

Energy generated per year = Average load × (365 × 24)

$= 78 \times 8760 = 683280 \times 10^3$ kWh

Coal required per year = $683280 \times 10^3 \times 1.2$

$= 819936000$ kg $= 819936$ tonnes

Cost of coal per year = $819936 \times 850 = \text{Rs. } 696.9456 \times 10^6$

Cost of energy sold = $683280 \times 10^3 \times 2$

$= \text{Rs. } 1366.56 \times 10^6$

(i) Revenue earned by the power plant per year

= cost of energy sold – cost of coal per year

$= \text{Rs. } 1366.56 \times 10^6 - 696.9456 \times 10^6$

$= \text{Rs. } 669.614 \times 10^6 = \mathbf{Rs.\ 66.96\ crore}$ ***Ans***

(ii) Capacity factor $= \dfrac{\text{Average load}}{\text{Capacity of plant}} = \dfrac{78}{170} = \mathbf{0.4588}$ ***Ans***

Problem 12.4 *A thermal power plant consists of two 50 MW units each running for 7600 hours and one 25 MW unit running for 2500 hours per year. The energy produced by the plant is 800×10^6 kWh/year. Determine plant load and plant use factor. Assume maximum demand is equal to plant capacity.*

Solution : Given Data :

Energy produces = 800×10^6 kWh/year, Load factor = ?, Plant use factor = ?

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Plant capacity = 50 + 50 + 25 = 125 MW

$$\text{Average load} = \frac{\text{Energy produced per year}}{365 \times 24} = \frac{800 \times 10^6}{8760}$$

$$= 91324.2 \text{ kW} = 91.324 \text{ MW}$$

$$\text{Load factor} = \frac{\text{Average load}}{\text{Peak load (Max. Load)}} = \frac{91.324}{125}$$

$$= 0.7306 = \mathbf{73.06\ \%}$$ ***Ans***

$$\text{Plant use factor} = \frac{\text{Actual energy produced}}{\text{Max. possible energy which can be produced by the plant}}$$

Max. possible energy which can be produced by the plant

$$= 2 \times 50 \times 7600 + 1 \times 25 \times 2500$$

$$= 822500 \text{ MWh} = 822.5 \times 10^6 \text{ kWh}$$

$$\therefore \text{ Plant use factor} = \frac{800 \times 10^6}{822.5 \times 10^6} = \mathbf{0.9726}$$ ***Ans***

Problem 12.5 *A diesel power plant consists of two units of 400 kW capacity each and one unit of 150 kW capacity. The fuel used has a calorific value of 42 MJ/kg and the fuel consumption is 0.23 kg/kWh. Determine the quantity of fuel required for a month of 30 days and its costs if the fuel cost is Rs. 1200 per tonne. Also find the overall efficiency of the plant. Take plant capacity factor on monthly basis = 55%.*

Solution : Given data :

C.V. = 42 MJ/kg, fuel consumption = 0.23 kg/kWh, quantity of fuel = ?, fuel cost = ?, η = ?

$$\text{Capacity factor} = \frac{E}{P_c \times t}$$

where E = energy generated during the month

P_c = plant capacity.

t = time in hours during the month

$\therefore\ P_c = 400 + 400 + 150 = 950$ kW.

$\therefore\ t = 30 \times 24 = 720\,h$

$$\therefore 0.55 = \frac{E}{950 \times 720} \quad \therefore E = 376200 \text{ kWh/month}$$

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Fuel consumption per month $= 0.23 \times 376200 = 86526\,kg$ = **86.525 tonnes** ***Ans***

Fuel cost = 86.526 × 1200 = **Rs. 103831.2/month** ***Ans***

$$\text{Overall efficiency} = \frac{\text{Output}}{\text{Input}} = \frac{950 \times 0.55}{\left(\frac{86526}{3600 \times 720}\right) \times 42 \times 10^3}$$

$$= 0.3727 = \mathbf{37.27\%}$$ ***Ans***

Problem 12.6 *An yearly load duration curve of a gas turbine power plant is straight line from 60000 kW to 8000 kW. The load is taken by a power plant which consists of two units of 25000 kW and one unit of 20000 kW. Determine (i) Load factor, (ii) capacity factor of the plant.*

Solution : Refer Fig. 12.15.

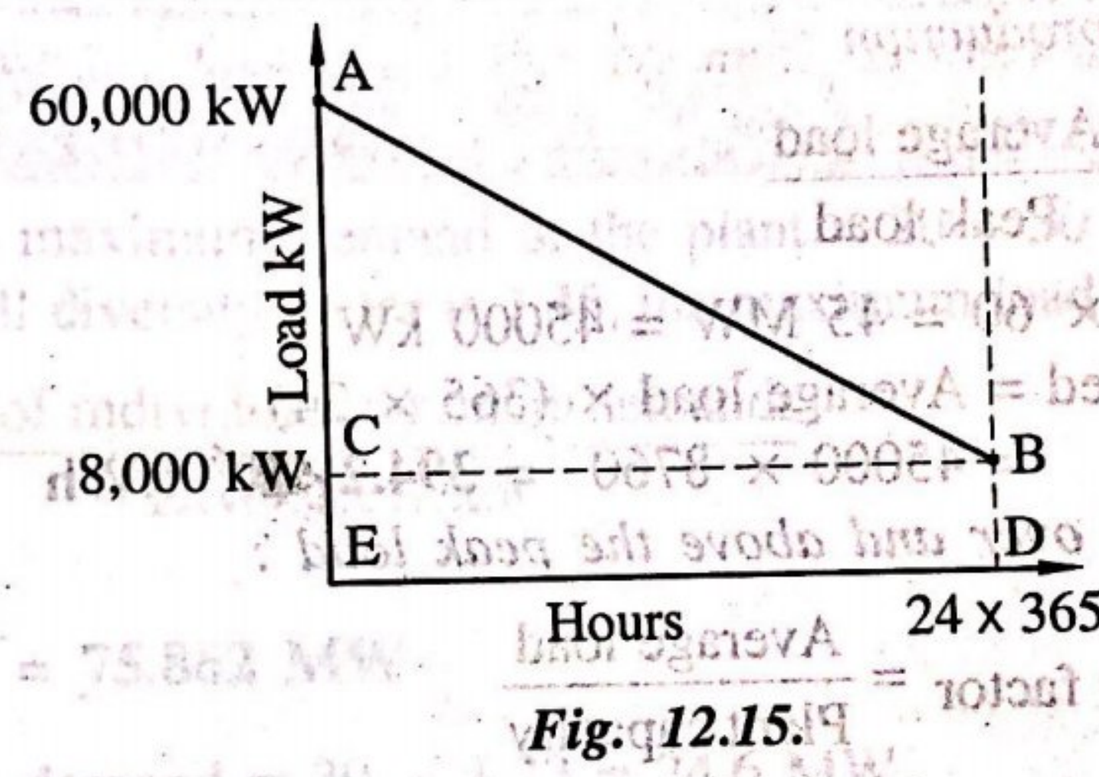


Fig. 12.15.

Total energy generated per year

= Area under the curve AB = Area (EDBC) + Area (ABC)

$$= 8000 \times 8760 + \frac{1}{2} \times 8760 \times (60{,}000 - 8000)$$

$$= 0.7008 \times 10^8 + 2.2776 \times 10^8$$

$$= 2.9784 \times 10^8 \text{ kWh}$$

$$\text{Average load} = \frac{2.9784 \times 10^8}{8760} = 34000 \text{ kW}$$

$$\text{(i) Load factor} = \frac{\text{Average load}}{\text{Maximum demand}} = \frac{34000}{60000} = 0.56667$$

$$= \mathbf{56.667\ \%}$$ ***Ans***

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(ii) Capacity factor $= \dfrac{\text{Energy generated/year}}{\text{Plant capacity} \times 8760}$

$$= \frac{2.9784 \times 10^8}{(25000 + 25000 + 20000) \times 8760}$$

$= 0.4857 = \mathbf{48.57\ \%}$ **Ans**

Problem 12.7 *A power plant has following annual factors : Load factor = 0.75, Capacity factor = 0.60, Maximum demand is = 60 MW. Estimate (a) The annual energy production, (b) The reserve capacity over and above the peak load and (c) The hours during which the plant not in service.* **[Dec. '10, 7 Marks]**

Solution : Given data

Load factor = 0.75, Capacity factor = 0.60, Maximum demand = 60 MW.

(a) The annual energy production :

We have, Load factor $= \dfrac{\text{Average load}}{\text{Peak load}}$

$\therefore$ Average load = $0.75 \times 60 = 45$ MW = 45000 kW

$\therefore$ Annual energy produced = Average load $\times$ (365 $\times$ 24)

$= 45000 \times 8760 = \mathbf{394.2 \times 10^6\ kWh}$ **Ans**

(b) The reserve capacity over and above the peak load :

We have, Annual capacity factor $= \dfrac{\text{Average load}}{\text{Plant capacity}}$

$\therefore$ Plant capacity $= \dfrac{45}{0.6} = 75$ MW

$\therefore$ Reserve capacity over and above peak load

= Plant capacity – Peak load = 75 – 60 = **15 MW** **Ans**

(c) The hours during which the plant not in service :

$$\text{Annual use factor} = \frac{\text{Energy generated per year}}{\text{Plant capacity} \times t_1}$$

where t_1 is the actual number of hours of the year for which the plant remains in operation.

$$\therefore\ 0.6 = \frac{394.2 \times 10^6}{75 \times 10^3 \times t_1} \quad (\because \text{ Assuming annual use factor} = 0.6)$$

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$$\therefore t_1 = \frac{394.2\times10^6}{75\times10^3\times0.6} = 8760 \text{ h}$$

$\therefore$ Hours not in service = $(365 \times 24) - t_1$

$= 8760 - 8760 = \mathbf{0h}$ ***Ans***

Problem 12.8 *The following loads are connected to a power plant :*

Type of load	*Max. demand (MW)*	*Diversity factor*	*Demand factor*
Domestic	*20*	*1.23*	*0.70*
Commercial	*30*	*1.20*	*0.88*
Industrial	*60*	*1.30*	*0.97*

If the overll diversity factor is 1.45, determine (i) the maximum load, and (ii) the connected load of each type.

Solution : The maximum demand of the plant = 20 + 30 + 60 = 110 MW

Since the overall diversity factor is 1.45, the maximum load of the system

$$= \frac{\text{Sum of individual maximum demands}}{\text{Diversity factor}}$$

$$= \frac{110}{1.45} = \mathbf{75.862\ MW}$$ ***Ans***

Max. domestic demand = $20 \times 1.23 = 24.6$ MW

$$\text{Connected domestic load} = \frac{24.6}{0.70} = 35.143 \text{ MW}$$

$$\text{Similarly, connected commercial load} = \frac{30\times1.20}{0.88} = 40.909 \text{ MW}$$

$$\text{Connected industrial load} = \frac{60\times1.30}{0.97} = 80.4123 \text{ MW}$$

$\therefore$ Total connected load of the plant = 35.143 + 40.909 + 80.4123

$= \mathbf{156.4643\ MW}$ ***Ans***

Problem 12.9 *It is proposed to supply a load with a maximum demand of 400 MW and load factor of 75%. Choice is to be made from a thermal power plant, a hydraulic power plant and a nuclear power plant. Calculate the overall cost per kWh in case of each scheme as given below :*

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Cost	Thermal Power plant	Hydraulic Power plant	Nuclear Power plant
1. Capital cost per MW installed	Rs. 5 crore	Rs. 7 crore	Rs. 8 crore
2. Interest	8%	6%	6%
3. Depreciation	7%	4%	6%
4. Operating cost (including fuel) per kWh	50 paise	8 paise	25 paise
5. Transmission and distribution cost per kWh	3 paise	5 paise	4 paise

Solution :

(i) Thermal power plant :

Capital cost = Rs. 5×400 crore = Rs. 2000×10^7

Interest = $0.08 \times 2000 \times 10^7$ = Rs. 160×10^7

Depreciation = $0.07 \times 2000 \times 10^7$ = Rs. 140×10^7

Annual fixed cost (interest + depreciation)

$= 160 \times 10^7 + 140 \times 10^7$ = Rs. 300×10^7

Energy generated per year = Average load $\times$ (24 $\times$ 365)

= Load factor $\times$ Maximum demand $\times$ 8760

$= 0.75 \times 400 \times 10^3 \times 8760$

$= 262.8 \times 10^7$ kWh

Running cost/kWh = (Operating cost/kWh) + (transmission and distribution cost/kWh)

= 50 + 3 = 53 paise

Overall cost/kWh = fixed cost/kWh + running cost/kWh

$$= \frac{\text{Rs.}300\times10^7}{262.8\times10^7} + 0.53 = \textbf{Rs. 1.6715}$$ ***Ans***

(ii) Hydraulic power plant :

Capital cost = Rs. 7×400 crore = Rs. 2800×10^7

Interest = $0.06 \times 2800 \times 10^7$ = Rs. 168×10^7

Depreciation = $0.04 \times 2800 \times 10^7$ = Rs. 112×10^7

Annual fixed cost = Interest + Depreciation

$= 168 \times 10^7 + 112 \times 10^7$ = Rs. 280×10^7

$$\textit{Fixed cost/kWh} = \frac{\text{Rs.}280\times10^7}{262.8\times10^7} = \text{Rs. } 1.065$$

Running cost/kWh = 8 + 5 = 13 paise

Overall cost/kWh = 1.065 + 0.13 = **Rs. 1.195**

Ans

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(iii) Nuclear power plant

Capital cost = Rs. 8×400 crore = Rs. 3200×10^7

Interest = $0.06 \times 3200 \times 10^7$ = Rs. 192×10^7

Depreciation = $0.06 \times 3200 \times 10^7$ = Rs. 192×10^7

Annual fixed cost = Rs. 384×10^7

$$\text{Fixed cost/kWh} = \frac{\text{Rs.}384\times10^7}{262.8\times10^7} = \text{Rs. } 1.461$$

Running cost/kWh = 25 + 4 = 29 paise

Overall cost/kWh = 1.461 + 0.29 = **Rs. 1.751** ***Ans***

Therefore, the overall cost/kWh is minimum in case of hydraulic power plant.

Problem 12.10 *A power plant of 300 MW installed capacity has the following particulars:*

Capital cost = Rs. 20000/kW installed

Interest and depreciation = 12%

Annual load factor = 65%

Annual capacity factor = 60%

Annual running charges = Rs. 290×10^6

Energy consumed by power plant auxiliary = 6%

Determine (i) the reserve capacity, and (ii) the cost of power generation per kWh

Solution : $\dfrac{\text{Load factor}}{\text{Capacity factor}} = \dfrac{\text{Average load}}{\text{Maximum demand}} \times \dfrac{\text{Capacity of the plant}}{\text{Average load}}$

$$\frac{0.65}{0.60} = \frac{300}{\text{Maximum demand}}$$

$\therefore$ Maximum demand = 276.923 MW

$\therefore$ Reserve capacity = Capacity of the plant – Maximum demand

= 300 – 276.923 = **23.076 MW** ***Ans***

Average load = Load factor × maximum demand

$= 0.65 \times 276.923 =$ 180 MW

Energy produced per year = Average load × (24 × 365)

$= 180\times10^3 \times 8760 = 1576.8\times10^6$ kWh

Net energy delivered $= 0.94\times1576.8\times10^6$

$= 1482.192 \times 10^6$ kWh

Annual interest and depreciation (fixed cost)

$= 0.12\times20000\times300\times10^3$ = Rs. 720×10^6

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Total annual cost = Fixed cost + Running cost

$$= 720\times10^6 + 290\times10^6 = \text{Rs.}\,1010\times10^6$$

$$\text{Cost of power generation} = \frac{\text{Rs.}\,1010\times10^6}{1482.192\times10^6\ kWh}$$

= Rs. 0.681 or 68.1 paise *Ans*

Problem 12.11 *Determine the cost of generation per kWh for a power station having the following data :*

Installed capacity of the plant = 300 MW
Capital cost = Rs. 600 crores
Rate of interest and depreciation = 12%
Annual cost of fuel, salaries and taxation = 8 crores
Load factor = 55%

Also estimate the saving in cost per kWh if the annual load factor is raised to 65%

[same as June 2012]

Solution : Assuming the maximum demand equal to the capacity of the power plant,

$$\text{Load factor} = \frac{\text{Average load}}{\text{Maximum demand}}$$

$$0.55 = \frac{\text{Average load}}{300}$$

$\therefore$ Average load = 165 MW

Energy generated per year = Average load × (365 × 24)

$$= 165\times10^3\times8760 = 144.54\times10^7\ \text{kWh}$$

Interest and depreciation (fixed cost)

$$= 0.12\times600\times10^7 = \text{Rs.}\,72\times10^7$$

$$\text{Total annual cost} = 72\times10^7 + 8\times10^7 = \text{Rs.}\,80\times10^7$$

$$\text{Cost per kWh} = \frac{\text{Rs.}\,80\times10^7}{144.54\times10^7\ \text{kWh}} = \textbf{Rs. 0.5534}$$ *Ans*

When the load factor is raised to 65%

Average load = 0.65×300 = 195 MW

Energy produced per year $= 195\times10^3\times8760$

$$= 170.82\times10^7\ \text{kWh}$$

Total annual cost will remain the same.

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$$\text{Cost per kWh} = \frac{\text{Rs. } 80\times10^7}{170.82\times10^7} = \text{Rs. } 0.468$$

$\therefore$ Saving in cost per kWh = 0.5534 – 0.468 = Rs. 0.0854 = **8.54 paise** ***Ans***

Problem 12.12 *A power station has to supply load as follows :*

Time (hours)	*0-6*	*6-12*	*12-14*	*14-18*	*18-24*
Load (MW)	*40*	*110*	*80*	*120*	*60*

(i) Draw the load curve, (ii) Draw the load duration curve. (iii) select suitable generating units to supply the load, (iv) calculate the load factor, (v) calculate the capacity of the plant and the plant capacity factor.

Solution : The load curve and load duration curve have been drawn in Fig. 12.16.

$$\text{Energy generated} = 40\times6+110\times6+80\times2+120\times4+60\times6$$

$$= 1900 \text{ MWh} = 1900\times10^3 \text{ kWh}$$

$$\text{Average load} = \frac{1900\times10^3}{24} kW$$

$$\text{Maximum demand} = 120\times10^3 \text{ kW}$$

$$\text{Load factor} = \frac{\text{Average load}}{\text{Maximum demand}}$$

$$= \frac{1900\times10^3}{24\times120\times10^3} = \mathbf{0.6597} \qquad \textit{Ans}$$

To supply the load two generating units each of 40 MW capacity and two generating units each of 20 MW capacity will be selected. One additional unit will be kept as standby. It capacity will be equal to 40 MW.

Load duration curve will indicate the operational schedule of different generating units, which will be as follows :

(a) One unit of 40 MW will run for 24 hours

(b) Second unit of 20 MW will run for 18 hours

(c) Third unit of 20 MW will run for 12 hours

(d) Fourth unit of 40 MW will run for 10 hours. ***Ans***

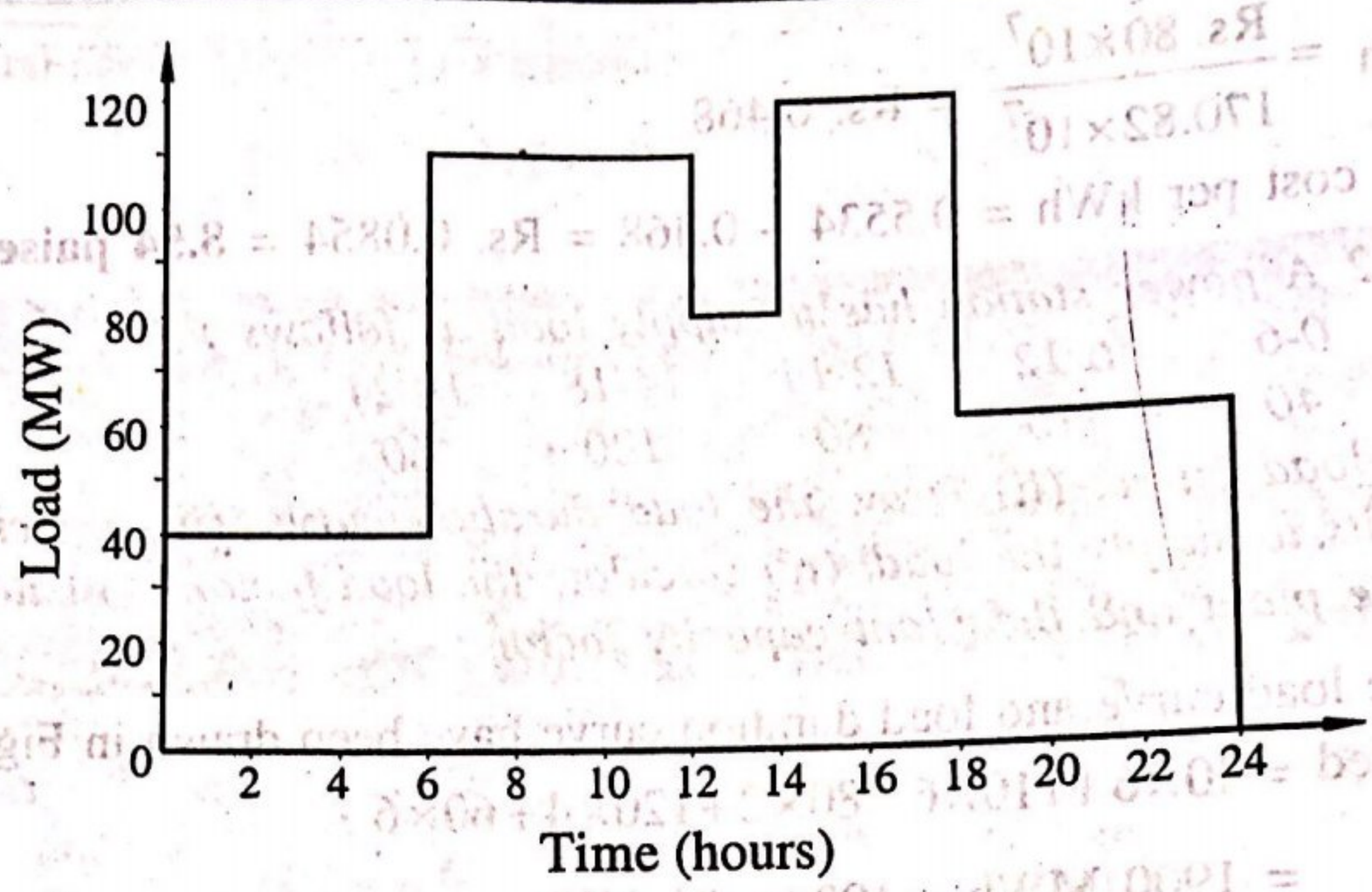
$$\text{Plant capacity} = 3\times40+2\times20 = 160 \text{ MW} \qquad \textit{Ans}$$

$$\text{Capacity factor} = \frac{\text{Energy generated}}{\text{Capacity}\times\text{operating time}}$$

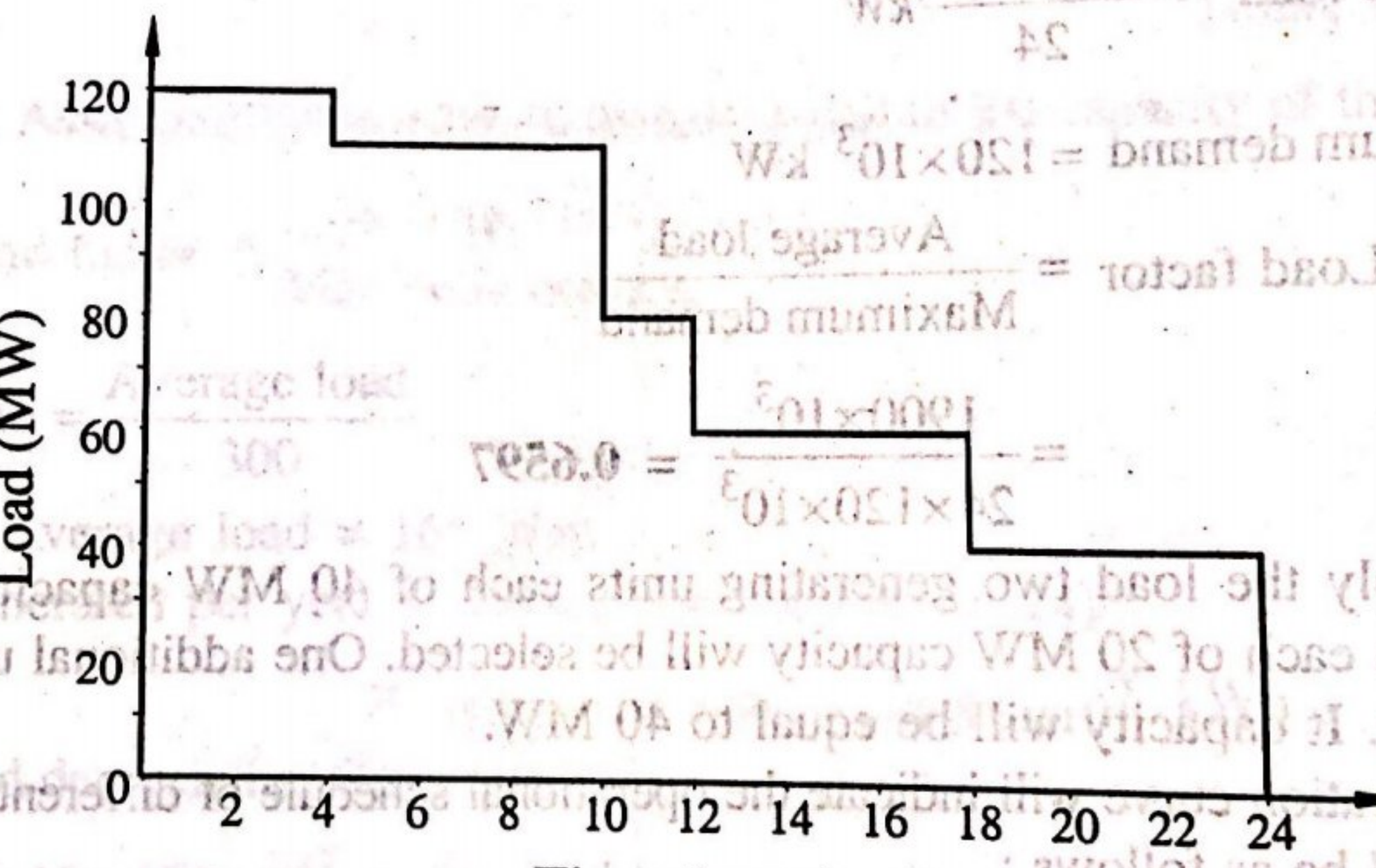
$$= \frac{1900\times10^3}{160\times10^3\times24} = \mathbf{0.4948} \qquad \textit{Ans}$$

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(a) Load curve



(b) Load duration curve

Fig. 12.16

Problem 12.13 *A generating unit of 20 MW capacity supplies the following loads:*
(a) Domestic consumers with a maximum demand of 12 MW at a load factor of 20%.
(b) Small industrial load with a maximum demand of 7 MW at a load factor of 50%.
(c) Street light load with a maximum demand of 1 MW at 30 % load factor.
Determine the overall cost of energy per kWh for each type of consumer using the following data :

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Capital cost of the plant = Rs. 12000 per kW
Total running cost = Rs. 73,00,000 per year
Annual rate of interest and depreciation on capital cost = 12%

Solution : Energy supplied per year to all three consumers

$$= [(12000 \times 0.2) + (7000 \times 0.5) + (1000 \times 0.3)] \times 8760$$
$$= 6200 \times 8760 \text{ kWh}$$

$$\text{Operating charges per kWh} = \frac{73,00,000}{6200 \times 8760} = \text{Rs. } 0.1344$$
$$= 13.44 \text{ paise}$$

$$\text{Capital cost of the plant} = 12000 \times 20 \times 10^3 = \text{Rs. } 2.4 \times 10^8$$

$$\text{Fixed charges per year} = 0.12 \times 2.4 \times 10^8 = \text{Rs. } 2.88 \times 10^7$$

$$\text{Fixed charges per kW} = \frac{2.88 \times 10^7}{20 \times 10^3} = \text{Rs. } 1440$$

(a) For domestic consumers,

Total charges = Fixed charge + operating charge

$$= 1440 \times 12 \times 10^3 + 0.1344 \times 12 \times 10^3 \times 0.2 \times 8760$$
$$= \text{Rs. } 20,105,625.6$$

$$\text{Overall cost per kWh} = \frac{\text{Rs. } 20,105,625.6}{0.2 \times 12 \times 10^3 \times 8760 \ kWh}$$

= Rs. 0.9563 or 95.63 paise ***Ans***

(b) For industrial consumers :

$$\text{Total charges} = 1440 \times 7 \times 10^3 + 0.1344 \times 7 \times 10^3 \times 0.5 \times 8760$$
$$= \text{Rs. } 14,200,704$$

$$\text{Overall cost per kWh} = \frac{\text{Rs. } 14,200,704}{0.5 \times 7 \times 10^3 \times 8760} = \textbf{Rs. 0.463 or 46.3 paise}$$ ***Ans***

(c) For street lighting load,

$$\text{Total charges} = 1440 \times 1000 + 0.1344 \times 1000 \times 0.3 \times 8760$$
$$= \text{Rs. } 1,793,203.2$$

$$\text{Overall cost per kWh} = \frac{\text{Rs. } 1,793,203.2}{0.3 \times 1000 \times 8760} = \textbf{Rs. 0.6823 or 68.23 paise}$$ ***Ans***

Problem 12.14 *A 80 MW turbo-generator set has an overall efficiency of 30%. The calorific value of coal used is 26000 kJ/kg. Find the consumption of coal for kWh and per day of 24 hours if the load factor is 40%.*

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Solution : Overall efficiency, $\eta = \dfrac{\text{Output}}{\text{Input}}$

If output is 1 kWh, then

$$\text{Input} = \frac{1}{0.30} = 3.333 \text{ kWh}$$

$$\therefore \text{ Consumption of coal per kWh} = \frac{3.333 \times 3600}{26000} = \mathbf{0.4615 \text{ kg}} \quad \textit{Ans}$$

$$\text{Load factor} = \frac{\text{Energy consumed in 24 hours}}{\text{Peak load} \times 24}$$

$\therefore$ Energy consumed in kWh within 24 hours

$$= 0.4 \times 80 \times 10^3 \times 24 = 768 \times 10^3 \ kWh$$

$\therefore$ Consumption of coal per day $= 768 \times 10^3 \times 0.4615 = 354.432 \times 10^3$ kg

$$= \mathbf{354.432 \text{ tons}} \quad \textit{Ans}$$

Problem 12.15 *The cost of water softner plant used is Rs. 2.8×10^5 when newly installed. The life of the plant is considered as 15 years. The salvage value of the plant will be 8% of its cost when newly installed. The repair, maintenance and labour costs of the plant per year are Rs. 12,000. The cost of chemical used per year is Rs. 10,000. Taking interest on sinking fund as 8%, find the annual cost of the plant.*

Solution : Given data :

$$P = \text{Rs. } 2.8 \times 10^5, \ S = \frac{8}{100} \times 2.8 \times 10^5 = \text{Rs. } 22400, \ r = 8\% = 0.08, \ n = 15,$$

Annual cost of the plant = ?

Annual sinking fund payment for the plant is given by

$$A = (P - S)\left[\frac{r}{(1+r)^n - 1}\right] = (2.8 \times 10^5 - 22400)\left[\frac{0.08}{(1+0.08)^{15} - 1}\right]$$

$$= 257600\left[\frac{0.08}{2.172}\right] = 9488.03$$

$\therefore$ Annual cost of the plant is given by

= Annual sinking fund payment + Repair, maintenance, and labour costs + chemical cost

$$= 9488.03 + 12000 + 10000 = \mathbf{Rs. \ 31488.03} \quad \textit{Ans}$$

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Problem 12.16 *An industrial power station has a maximum demand of 100 MW and installed capacity is a single generator of 70 MW. Excess energy if required is brought from the state grid. The load curve of the plant is defined as follows.*

Time (hours)	*0-6*	*6-8*	*8-12*	*12-14*	*14-18*	*18-22*	*22-24*
Load (MW)	*40*	*50*	*80*	*50*	*90*	*100*	*45*

Determine (i) load factor of power station, (ii) load factor of generator, (iii) energy purchased from grid per day, (iv) plant use factor. ***[same as June '11, 7 marks]***

Solution : Given data

Generator capacity = 70 MW, Maximum demand = 100 MW

The load curve for given data is shown in Fig. 12.17.

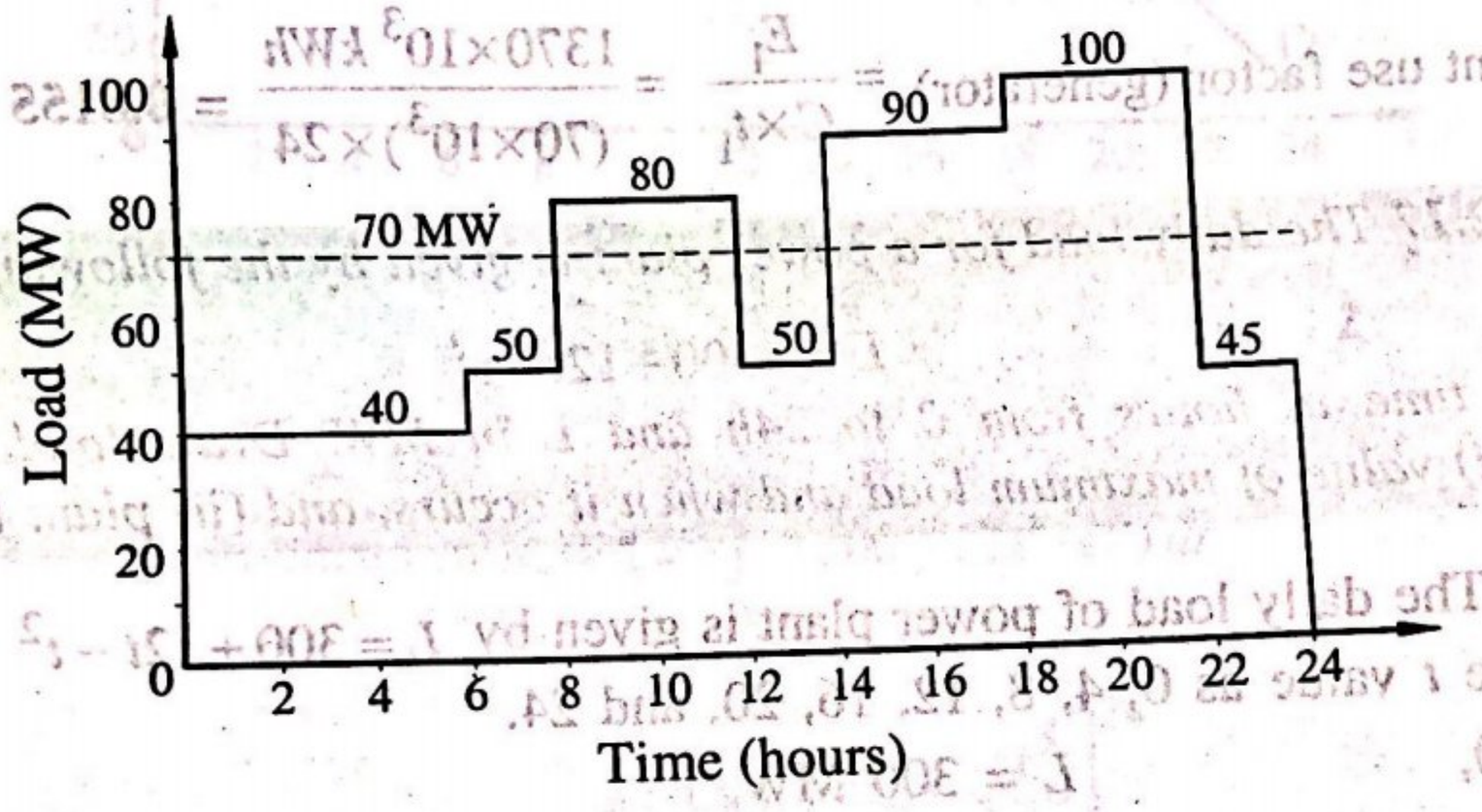


Fig. 12.17

(i) Load factor for power station

Energy generated per day,

$$E = 40\times6+50\times2+80\times4+50\times2+90\times4+100\times4+45\times2$$

$$= 1610 \text{ MWh} = 1610\times10^3 \text{ kWh}$$

$$\text{Average load} = \frac{1610}{24} = 67.083 \text{ MW}$$

$$\therefore \text{Load factor} = \frac{\text{Average load}}{\text{Maximum demand}} = \frac{67.083}{100} = \mathbf{0.67083} \quad \textbf{Ans}$$

(ii) Load factor for generator

Energy produced by generator,

$$E_1 = \text{Area under the dotted curve}$$

$$= 40\times6+50\times2+70\times4+50\times2+70\times4+70\times4+45\times2$$

$$= 1370 \text{ MWh}$$

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Average load on generator $= \frac{1370}{24} = 57.0833$ MW

Maximum demand on generator = 70 MW.

$\therefore$ Load factor for generator $= \frac{57.0833}{70} = \mathbf{0.8155}$ ***Ans***

(iii) Energy purchased from grid per day, ΔE;

$$\Delta E = E - E_1 = 1610 - 1370 = 240 \text{ MWh}$$

$$= \mathbf{240 \times 10^3 \text{ kWh}}$$ ***Ans***

(iv) Plant use factor

$$\text{Plant use factor (generator)} = \frac{E_1}{C \times t_1} = \frac{1370 \times 10^3 \text{ kWh}}{(70 \times 10^3) \times 24} = \mathbf{0.8155}$$ ***Ans***

Problem 12.17 *The daily load for a power plant is given by the following equation.*

$$L = 300 + 12t - t^2$$

where t is time in hours from 0 to 24h and L in MW. Draw load curve and calculate: (i) value of maximum load and when it occurs, and (ii) plant load factor.

Solution : The daily load of power plant is given by $L = 300 + 12t - t^2$

We will take t value as 0, 4, 8, 12, 16, 20, and 24.

$t = 0$, $L = 300$ MW

$t = 4$, $L = 300 + 12 \times 4 - 4^2 = 332$ MW

$t = 8$, $L = 300 + 12 \times 8 - 8^2 = 332$ MW

$t = 12$, $L = 300 + 12 \times 12 - 12^2 = 300$ MW

$t = 16$, $L = 300 + 12 \times 16 - 16^2 = 236$ MW

$t = 20$, $L = 300 + 12 \times 20 - 20^2 = 140$ MW

$t = 24$, $L = 300 + 12 \times 24 - 24^2 = 12$ MW

The load curve can be plotted as shown in Fig. 12.18.

(i) Value of maximum load,

Condition for maximum load point is given by

$$\frac{d(L)}{dt} = 0; \quad \frac{d(L)}{dt} = 12 - 2t = 0$$

$\therefore$ $t = 6$ *h*

Ans

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$\therefore$ At $t = 6\ h$, the load is maximum

$\therefore L_{max} = 300 + 12 \times 6 - 6^2 =$ **336 MW** ***Ans***

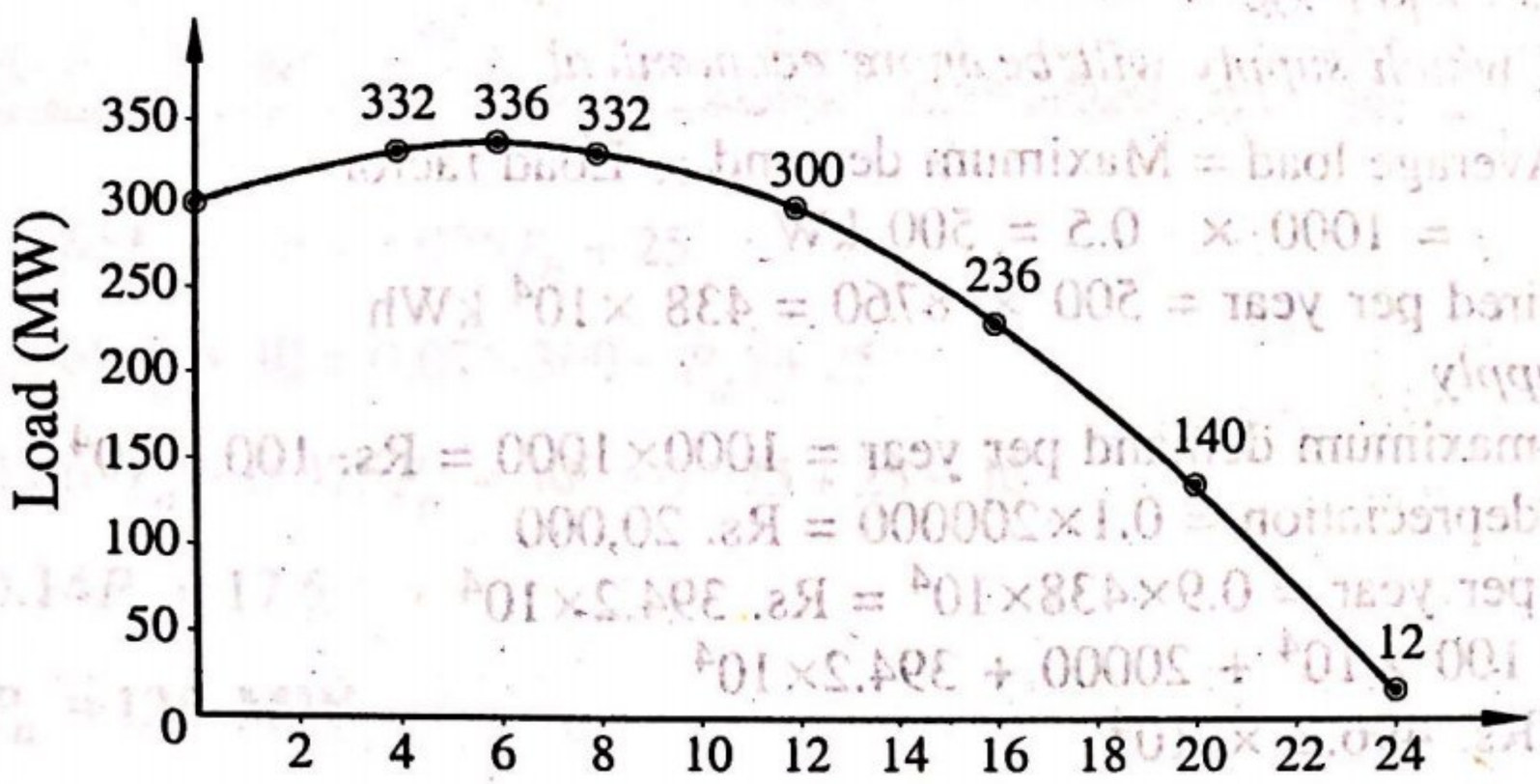


Fig. 12.18

(ii) Plant load factor

$$\text{Total energy produced in MWh} = \int_0^{24} (300 + 12t - t^2)\, dt$$

$$= \left[300t + \frac{12}{2}t^2 - \frac{t^3}{3}\right]_0^{24}$$

$$= 300 \times 24 + 6(24^2) - \frac{1}{3}(24^3)$$

$$= 6048 \text{ MWh}$$

$$\text{Average load} = \frac{6048}{24} = 252 \text{ MW}$$

$$\text{Plant load factor} = \frac{\text{Average load}}{\text{Maximum load}} = \frac{252}{336} = \mathbf{0.75}$$ ***Ans***

Problem 12.18 *A new industry requires a maximum demand of 1000 kW at 50% load factor. The following two power suppliers are available.*
(i) Public supply charges Rs. 1000/kW of maximum demand and 90 paise per kWh consumed. The capital cost is Rs. 2,00,000 and interest and depreciation charges on capital are 10%.

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(ii) A private oil engine station requires capital of Rs. 2500 per kW. The interest and depreciation on the capital are 12%. The maintenance and labour charges are 5 paise per kWh energy generated. The fuel consumption is 0.3 kg per kWh and cost of fuel is Rs. 3 per kg.

Find which supply will be more economical ?

Solution : Average load = Maximum demand × Load factor

$= 1000 \times 0.5 = 500$ kW

Energy required per year = $500 \times 8760 = 438 \times 10^4$ kWh

(i) Public supply

Charges for maximum demand per year = 1000×1000 = Rs. 100×10^4

Interest and depreciation = 0.1×200000 = Rs. 20,000

Energy cost per year = $0.9 \times 438 \times 10^4$ = Rs. 394.2×10^4

Total cost = $100 \times 10^4 + 20000 + 394.2 \times 10^4$

= Rs. 496.2×10^4

$$\text{Average energy cost} = \frac{496.2 \times 10^4}{438 \times 10^4} = \text{Rs. } 1.1328/\text{kWh}$$

(ii) Private supply

$$\text{Full consumption per year} = \frac{0.3 \times 438 \times 10^4}{1000} = 1314 \text{ tons}$$

Fuel cost = $1314 \times 1000 \times 3$ = Rs. 3942000

The maintenance and labour charges per year = $0.05 \times 438 \times 10^4$

= Rs. 219000

Interest and depreciation = $0.12 \times 2500 \times 1000$ = Rs. 3,00,000

Total cost = 3942000 + 219000 + 300000 = Rs. 4461000

$$\therefore \text{ Average energy cost} = \frac{4461000}{438 \times 10^4} = \text{Rs. } 1.0185$$

As the average energy cost for oil engine is less than the public supply, the oil engine generation is more preferable.

Problem 12.19 *Two power plants A and B, have incremental fuel costs given by* $\frac{dF_a}{dP_a} = 0.065P_a + 30$ *and* $\frac{dF_b}{dP_b} = 0.075P_b + 25$ *where F is fuel cost in rupees per hour and P is power output in MW. If the total power supplied by both the units is 300 MW, find the economic loading for the two units.*

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Solution : $P_a + P_b = 300$ MW, $P_b = 300 - P_a$.

For economic loading of plant, the condition is that

$$\frac{dF_a}{dP_a} = \frac{dF_b}{dP_b}$$

$$\therefore 0.065P_a + 30 = 0.075P_b + 25$$

$$\therefore 0.065P_a + 30 = 0.075(300 - P_a) + 25$$

$$\therefore 0.065P_a + 0.075P_a = 300 \times 0.075 + 25 - 30$$

$$\therefore 0.14P_a = 17.5$$

$$\therefore P_a = \mathbf{125\ MW} \qquad \textit{Ans}$$

$$P_b = 300 - 125 = \mathbf{175\ MW} \qquad \textit{Ans}$$

Problem 12.20 *The incremental fuel costs for two generating units a and b of a power plant are given by the following relations.*

$$\frac{dF_a}{dP_a} = 0.06P_a + 25 \quad \textit{and} \quad \frac{dF_b}{dP_b} = 0.075P_b + 20$$

where F is the fuel cost in rupees per hour and P is the power output in MW. Find (i) the economic loading of the two units when the total load supplied by the power plants is 300 MW, (ii) the loss in fuel cost per hour if the load is equally shared by both units.

Solution $P_a + P_b = 300$ MW, $P_b = 300 - P_a$

For economic loading of plant, the condition is that

$$\frac{dF_a}{dP_a} = \frac{dF_b}{dP_b}$$

$$\therefore 0.06P_a + 25 = 0.075P_b + 20$$

$$\therefore 0.06P_a + 25 = 0.075(300 - P_a) + 20$$

$$\therefore 0.06P_a + 0.075P_a = 0.075 \times 300 + 20 - 25$$

$$\therefore 0.135P_a = 17.5; \quad P_a = \mathbf{129.63\ MW} \qquad \textit{Ans}$$

$$P_b = 300 - 129.63 = \mathbf{170.37\ MW} \qquad \textit{Ans}$$

(ii) If the load is equally shared by the units, i.e. $P_a = P_b = 150$ MW, then the increase in cost of fuel for unit *a* is given by

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$$\int \frac{dF_a}{dP_a} dP_a = \int_{129.63}^{150} (0.06P_a + 25)dP_a$$

$$= \left[0.06 \frac{P_a^2}{2} + 25P_a \right]_{129.63}^{150}$$

$$= \left[\frac{0.06}{2} (150^2 - 129.63^2) + 25(150 - 129.63) \right]$$

$$= 680.132 \quad \text{i.e. Rs. } 680.132 \text{ per hour}$$

The increase in cost for unit b is

$$= \int_{170.37}^{150} (0.075P_b + 20)dP_b$$

$$= \left[0.075 \frac{P_b^2}{2} + 20P_b \right]_{170.37}^{150} = -652.1226 \text{ per hour} \qquad \textit{(ii)}$$

(1)

that is, the cost of fuel for unit b decreases by Rs. 652.1226 per hour. Therefore, the net increase in fuel cost due to departure from economic distribution of load.

$$= 680.132 - 652.1226 = \textbf{Rs. 28.01 per hour} \qquad \textit{Ans}$$

Problem 12.21 *A 10 MW capacity power plant has the input-output curve defined by the equation :*

$$I = 4 \times 10^6 (8 + 6L + 0.5L^2) \text{ kJ/h}$$

where L is in MW. Determine (i) load at which the plant has its maximum efficiency and its value, (ii) increase in input required to increase the output from 5 MW to 7 MW by using the input-output equation and also by incremental rate curve.

Solution : *(i) Load at which efficiency,* η *is maximum*

$$\eta = \frac{L}{I} = \frac{L}{4 \times 10^6 (8 + 6L + 0.5L^2)}$$

$$= \frac{1}{4 \times 10^6 \left(\frac{8}{L} + 6 + 0.5L \right)}$$

The efficiency is maximum when $\left(\frac{8}{L} + 6 + 0.5L \right)$ is minimum, therefore,

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$$\frac{d}{dL}\left(\frac{8}{L}+6+0.5L\right)=0$$

$$-\frac{8}{L^2}+0.5=0;\quad -8+0.5L^2=0$$

$$\therefore L^2=\frac{8}{0.5}=16$$

$$\therefore L = 4 \text{ MW} = \mathbf{4000\ KW}$$

$$=4000\times3600\,\text{kJ/h} = \mathbf{14.4\times10^6\ kJ/h} \quad \textit{Ans}$$

$$\therefore I=4\times10^6(8+6\times4+0.5\times4^2)$$

$$=160\times10^6\ \text{kJ/h}$$

$$\eta_{max}=\frac{L}{I}=\frac{14.4\times10^6}{160\times10^6}=\mathbf{0.09\ or\ 9\%} \quad \textit{Ans}$$

(ii) Increase in input when L_1 = 5 MW is increased to L_2 = 7 MW.

(1) By input-output curve method :

$$I_2-I_1=4\times10^6[8+6L_2+0.5L_2^{\,2}]-4\times10^6[8+6L_1+0.5L_1^{\,2}]$$

$$=4\times10^6[6(L_2-L_1)+0.5(L_2^{\,2}-L_1^{\,2})]$$

$$=4\times10^6[6(7-5)+0.5(7^2-5^2)]$$

$$=\mathbf{96\times10^6\ kJ/h} \quad \textit{Ans}$$

(2) By increment heat rate method :

$$IR=\frac{dI}{dL}=\frac{d}{dL}[4\times10^6(8+6L+0.5L^2)]$$

Average load, $L=\frac{7+5}{2}=6$ MW

$$\therefore IR=4\times10^6(6+1\times6)=48\times10^6\ \text{kJ/h}$$

$$\therefore \text{Total increase in input} = IR\times(L_2-L_1)$$

$$=48\times10^6\times(7-5)$$

$$=\mathbf{96\times10^6\ kJ/h} \quad \textit{Ans}$$

This shows that increase in input required to increase the given output in both

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cases (1) and (2) above is same. It shows the IR curve can be taken as straight line for small increase in load.

Problem 12.22 *The input-output curve of a 50 MW power station is given by*

$$I = 2\times10^6[6+6L+0.5L^2] \text{ kJ/h}$$

where I is the input in kJ/h and L is load in MW.

(i) Determine the heat input per day to the power station if it works for 18 hours at full load and remaining period at no load.

(ii) Also find the saving per kWh of energy produced if the plant works at full load for all 24 hours generating the same amount of energy.

Solution :

(i) Heat input per day

The plant works for $t_1 = 18\,\text{h}$ at $L = 50$ MW and

$t_2 = (24-18) = 6h$ per day, $L = 0$ MW (no load)

Total energy generated, $E = 18\times50+6\times0 = 900$ MWh

$$= 900\times10^3 \text{ kWh}$$

Total input, $I = (I_{50})\times t_1 + I_0 \times t_2$ kJ/day

$$= 2\times10^6[6+6\times50+0.5\times50^2]\times18+2\times10^6[6+6\times0+0.5\times0^2]\times6$$

$$= 56016\times10^6 + 72\times10^6$$

$$I = \mathbf{56088 \times 10^6 \text{ kJ/day}} \qquad \textit{Ans}$$

(ii) Saving in input per kWh :

Average heat input per kWh of energy generated

$$= \frac{I}{E} = \frac{56088\times10^6}{900\times10^3} = 62320 \text{ kJ/kwh}$$

If the same energy is generated within 24 h, the average load is given by $(t_3 = 24h)$

$$\text{Average load} = \frac{900}{24} = 37.5 \text{ MW}$$

Heat supplied per day at this load will be

$$I_1 = I_{37.5}\times t_3 = 2\times10^6[6+6\times37.5+0.5\times(37.5)^2]\times24$$

$$= 44838\times10^6 \text{ kJ/day}$$

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Net saving per day $= I - I_1 = 56088\times10^6 - 44838\times10^6$

$= 11250\times10^6$ kJ/day

$\therefore$ Saving in input per kWh $= \dfrac{I - I_1}{E} = \dfrac{11250\times10^6}{900\times10^3} = \mathbf{12.5\times10^3}$ **kJ/kWh** ***Ans***

Problem 12.23 *Two power plants are compared for their economical working having the details as follows.*

(i) Cost of first power plant is Rs. 15000 and it takes 90 kW of load.

(ii) Cost of second power plant is Rs. 30000 and it takes 60 kW of load. Each of them has a useful life of 80,000 hours. Which unit will prove economical if the energy is charged at Rs. 150 per kW of maximum demand per year and 20 paise per kWh ? Assume both units run at full load.

Solution :

(i) First power plant :

Capital cost per hour $= \dfrac{\text{Cost of plant}}{\text{Life in hour}} = \dfrac{15000}{80000} =$ Rs. 0.1875

Maximum demand = 90 kW

Charge for maximum demand per hour

$= \dfrac{\text{Maximum demand} \times \text{Cost/kW of demand}}{\text{Number of hours per year (8760 h)}} = \dfrac{90\times150}{8760}$

= Rs. 1.54

Energy charge per hour = Maximum demand × one hour × charge/kWh

$= 90\times1\times0.2 =$ Rs. 18

Total charges per hour of first plant

$= 0.1875 + 1.54 + 18 =$ **Rs. 19.7275**

(ii) Second power plant :

Capital cost per hour $= \dfrac{30000}{80000} =$ Rs. 0.375

Maximum demand = 60 kW

Charge for maximum demand per hour $= \dfrac{60\times150}{8760} =$ Rs. 1.0274

Energy charge per hour $= 60\times1\times0.2 =$ Rs. 12

Total energy charge per hour = 0.375 + 1.0274 + 12 = **Rs. 13.4024**

The charges of operation for the second plant per hour are less than the charges of operation for the first plant, therefore the second unit is more economical in this case.

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Problem 12.24 *The energy consumption of a consumer per month is 3000 kWh. The maximum demand is 14 kW. Using the Hopkinson demand rate as given below, find (i) monthly bill of the consumer and unit energy cost, (ii) lowest possible bill for a month of 30 day and unit energy cost for the given energy consumption.*

The Hopkinson charges are as follows :

0-5 kW = Rs. 80 per kW
6-10 kW = Rs. 60 per kW
10-15 kW = Rs. 40 per kW
First 200 kWh = Rs. 1.0/kWh
Next 1000 kWh = Rs. 0.9/kWh
Next 2000 kWh = Rs. 0.8/kWh
Excess over 2000 kWh = Rs. 0.6/kWh

Solution :

(i) Monthly bill of the consumer :

Demand charge per month $= 5\times80 + 5\times60 + 4\times40 =$ Rs. 860

Energy charge = Rs $[200\times1 + 1000\times0.9 + 1800\times0.8]$ = Rs. 2540

$\therefore$ Monthly bill = 860 + 2540 = **Rs. 3400** ***Ans***

Average unit energy cost $= \dfrac{3400}{3000} =$ **Rs. 1.1333/kWh** ***Ans***

(ii) The lowest bill occurs when the demand is maximum which is only possible a 100% load.

$\therefore$ Maximum load = Average load

$$= \frac{\text{Energy}}{\text{time}} = \frac{3000}{30\times24} \quad (\because 30 \text{ days} = 1 \text{ month})$$

$$= 4.1667 \text{ kW}$$

$\therefore$ Demand charges = $4.1667 \times 80 =$ Rs. 333.336

Energy charges will be same as Rs. 2540

$\therefore$ Minimum monthly bill = 333.336 + 2540 = Rs. 2873.336

Unit energy charge for this condition $= \dfrac{2873.336}{3000} =$ **Rs. 0.9578/kWh** ***Ans***

Problem 12.25 *Determine the load factor at which the cost of supplying a unit of electricity is same in Diesel power station as in a steam power station if the respective annual fixed and running charges are given below.*

Steam power plant : Rs. (200/kW + 0.08/kWh)
Diesel power plant : Rs. (70/kW + 0.1/kWh)

Solution : Consider both diesel and steam power plant and let,

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P = Maximum load in kW

y = Load factor

Average load = $P.y$

Cost of steam power plant,

$$C_s = Rs.(200 \times P + 0.08 \times P \times y \times 8760)$$

Cost of diesel power plant,

$$C_d = Rs.(70 \times P + 0.1 \times P \times y \times 8760)$$

where 8760 represents the number of hours run per year.

As given in the problem.

Unit energy cost by Diesel = Unit energy cost by steam

$$\frac{C_d}{P \times y \times 8760} = \frac{C_s}{P \times y \times 8760}$$

$$\therefore \frac{70P + 0.1Py \times 8760}{P \times y \times 8760} = \frac{200P + 0.08Py \times 8760}{P \times y \times 8760}$$

$$\therefore 70P + 876Py = 200P + 700.8Py$$

$$\therefore 175.2Py = 130P$$

$$\therefore y = \frac{130}{175.2} = \mathbf{0.742}$$ **Ans**

Exercises

12.1 Explain briefly the following :
(i) Load curve, (ii) Load duration curve.

12.2 What is the significance of load curves ?

12.3 Define : Load factor, Diversity factor, and plant use factor.

12.4 What do you understand by load factor and capacity factor ? When are they numerically equal ?

12.5 How is the load duration curve constructed ?

12.6 Define the following terms :
(i) Peak load, (ii) Average load, (iii) Plant capacity factor, (iv) Connected load.

12.7 Discuss the effect of variation of load factor and diversity factor on the design of a power plant.

12.8 What do you understand by base load and peak load power plants ?

12.9 Explain the method by which economic load sharing between base load and peak load plants can be determined.

12.10 Discuss the methods of meeting variable loads on power plant.

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12.11 List the various costs which go to form the total cost of a power system.

12.12 Explain briefly the following :
(i) Capital or fixed cost,, (ii) Operational cost.

12.13 What do you mean by depreciation ?

12.14 Enumerate and explain briefly various methods used to calculate the depreciation cost.

12.15 Name the elements that make up the operating expenditure of a power plant.

12.16 What do you mean by the sinking-fund method of accumulating the money for the depreciation fund ?

12.17 What is the significance of incremental rate for a power plant ?

12.18 When does heat rate equal the incremental rate ?

12.19 Keeping in view the operating and fixed costs of a power station, suggest some suitable tariffs.

12.20 What is the significance of two part tariff and three part tariff ?

12.21 What do you understand by the term tariff ?

Unsolved Numerical Problems

12.1 A power station supplies the following loads to the consumers :

Time in hours	0-6	6-8	8-12	12-14	14-18	18-22	22-24
Load in MW	40	50	60	50	70	80	40

(i) Draw the load curve and estimate the load factor of the plant, (ii) what is the load factor of a standby equipment of 25 MW capacity if it takes up all loads above 60 MW ? What is its use factor ?

[**Ans :** (i) Load factor = 0.71, (ii) Load factor = 0.75, use factor = 0.6]

12.2 The annual peak load on a 60 MW power station is 50 MW. The power station supplies load having maximum demand of 20 MW, 17 MW, 10 MW and 9 MW. The annual load factor is 0.45. Find (i) average load, (ii) energy supplied per year, (iii) diversity factor, (iv) demand factor.

[**Ans :** (i) 22.5 MW, (ii) 197.1×10^6 kWh, (iii) 1.12, (iv) 0.89]

12.3 An yearly load duration curve of a power plant is straight line from 30,000 kW to 4,000 kW. To meet the load three turbo-generator are installed. The capacity of two generator is 15,000 kW each and the third is rated at 5,000 kW. Determine (i) load factor, (ii) capacity factor, (iii) maximum demand.

[**Ans :** (i) 0.57, (ii) 0.49, (iii) 30,000 kW]

12.4 A power plant has following annual factors :
Load factor = 70%, capacity factor = 60%, and use factor = 65%. The power plant has a maximum demand of 50 MW. Estimate (i) annual energy production, (ii) reserve capacity above peak load, (iii) hours per year the plant is not in use.

[**Ans :** (i) 306.6×10^6 kWh, (ii) 8333.33 kW, (iii) 673.85h]

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12.5 A power plant of 200 MW installed capacity has the following particulars :
Capital cost of the plant = Rs. 140×10^6
Interest and depreciation = 11% of the capital
Annual load factor = 65%
Annual capacity factor = 55%
Annual cost of coal, oil, taxes and salaries = Rs. 21×10^6
Energy consumed by power plant auxiliaries = 4%
Determine : (i) reserve capacity, (ii) generating cost per kWh
[**Ans :** (i) 962×10^6 kWh, (ii) 3.64 paise per kWh]

12.6 A generating unit of 120 MW capacity supplies the following loads.
(a) Domestic consumers with a maximum demand of 75 MW at a load factor of 30%.
(b) Small industrial load with a maximum demand of 70 MW at a load factor of 50%
(c) Street-light load with a maximum demand of 5 MW at a load factor of 20%
Determine the overall cost of energy per kWh for each type of consumer using the following data :
Capital cost of the plant = Rs. 30,000 per kW, Annual rate of interest and depreciation on capital = 12%, Total running cost = Rs. 20 crores per year.
[**Ans :** (a) Rs. 1.91, (b) Rs. 0.55, (c) Rs. 2.60]

12.7 A new industry requires a maximum demand of 1 MW at 40% load factor. The following two power suppliers are available :
(i) State electricity board supply which charges Rs. 500/kW of maximum demand and maintenance and labour charges of 10 paise per kWh. The capital cost is Rs. 150,000 and interest and depreciation charges on capital are 10%.
(ii) A private gas turbine power plant requires Rs. 3 crores of capital. Interest and depreciation on the capital are 12%. Maintenance and labour charges are 5 paise per kWh of energy generated. The fuel consumption is 0.35 kg/kWh and cost of fuel is Rs. 6 per kg. Determine which supply is more economical.
[**Ans :** (i) 24.7 paise/kWh, (ii) Rs. 3.18/kWh, State electricity board]

12.8 Determine cost of power generation per kWh for a power station having the following data :
Installed capacity of the plant = 120 MW
Capital cost = Rs. 96×10^6
Rate of interest and depreciation = 14%
Annual cost of fuel oil, salaries and taxation = Rs. 12×10^5
Load factor = 40%
Also estimate the saving in cost per kWh if the annual load factor is raised to 50%. [**Ans :** cost/kWh = 6 paise, saving in cost/kWh = 1.18 paise]

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12.9 The input-output curve of a 10 MW power station is given by :

$$I = 4\times10^6[10+8L+0.4L^2]kJ/h$$

where I is the input in kJ/h and L is load in MW

(i) Find the load at which the maximum efficiency occurs.

(ii) Find the increase in input required to increase station output form 3 to 5 MW by means of the input-output curve and also by incremental rate curve.

[**Ans :** (i) 5 MW, (ii) 89.6×10^6 kJ/h]

12.10 The input-output curve of a 50 MW power station is given by :

$I = 4\times10^6[8+8L+0.4L^2]$ where I is the input in kJ/h and L is load in MW.

(i) Calculate the heat input per day to the power plant if it works for 20 hours at full load and remaining period at no load.

(ii) Also determine the saving per kWh of energy produced if the plant works at full load for all 24 hours generating the same amount of energy.

[**Ans :** (i) 112768×10^6 kJ/day (ii) 6928 kJ/kWh]

12.11 Two power plants are compared for their economical working having the details as follows :

(i) cost of first unit = Rs. 7000 and it takes 90 kW

(ii) cost of second unit = Rs. 18000 and it takes 50 kW

Each of them has useful life of 38000 hours. Which unit will prove economical if the energy is charged at Rs. 100 per kW of maximum demand per year and 8 paise per kWh ? [**Ans** : (i) Rs. 8.412/h, (ii) Rs. 5.04/h second unit]

12.12 The incremental fuel costs for two generating units a and b of a power plant are given by the following equations

$$\frac{dF_a}{dP_a} = 0.07P_a + 24 \text{ and } \frac{dF_b}{dP_b} = 0.075P_b + 22$$

where F is fuel cost in rupees per hour and P is power output in MW. Calculate (i) the economic loading of the two units when the total load supplied by the power plant is 180 MW. (ii) the loss in fuel cost per hour if the load is equally shared by both units. [**Ans:**(i) P_a = 79.3 MW, P_b = 100.7 MW, (ii) Rs. 8.3/h]

Objective Type Questions

1. The power plant design is made on the basis of following condition
   (a) most efficient (b) most economical
   (c) near to load centre (d) none of the above
2. The load curves will not give following informations :
   (a) the energy generated in the period considered.

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(b) the maximum demand on the power plant.
(c) cost of energy generated
(d) it is useful to decide the economical sizes of various generating units.

3. Load factor of a power plant is defined as
(a) average load/plant capacity (b) plant capacity/average load
(c) maximum load/average load (d) average load/maximum load

4. Load factor of power plant is generally
(a) less than unity (b) equal to unity
(c) more than unity (d) none of the above

5. The ratio of the sum of the maximum demands of the individual consumers to the simultaneous maximum demand of the whole group during a particular time is called
(a) plant capacity factor (b) plant use factor
(c) diversity factor (d) demand factor

6. Diversity factor is always
(a) more than unity (b) equal to unity
(c) less than unity (d) none of the above

7. Higher value of load factor indicates that
(a) cost of power generation per unit is increased
(b) cost of power generation per unit is decreased
(c) total plant capacity is not properly utilised for most of the time
(d) all of the above

8. The ratio of maximum demand to connected load is called
(a) demand factor (b) plant use factor
(c) plant capacity factor (d) diversity factor

9. The load factor for domestic load is usually
(a) 10 to 15% (b) 30 to 40% (c) 50 to 60% (d) 60 to 70%

10. When will be the load factor and capacity factor numerically equal ?
(a) at maximum efficiency of the plant
(b) at the rated capacity of the plant is less than the peak load.
(c) at the rated capacity of the plant is more than the peak load.
(d) at the rated capacity of the plant is equal to the peak load.

11. Annual depreciation cost is calculated by
(a) straight line method (b) sinking fund method
(c) diminishing value method (d) all of the above

12. _____ is defined as the ratio of additional input required to increase the additional output.
(a) Heat rate (b) Incremental rate
(c) Demand rate (d) none of the above

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13. In two part tariff, variation in load factor will affect
(a) fixed and running charges (b) only fixed charges
(c) only running charges (d) none of the above

14. A consumer has to pay no fixed charges in
(a) flat rate tariff (b) two-part tariff
(c) straight line meter rate (d) three-part tariff

Answers

1. (b) **2.** (c) **3.** (d) **4.** (a) **5.** (c) **6.** (a) **7.** (b) **8.** (a) **9.** (a) **10.** (d)
11. (d) **12.** (b) **13.** (c) **14.** (c)

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